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MERADCOM/OSU HYDRAULIC SYSTEM RELIABILITY PROGRAM.(U)
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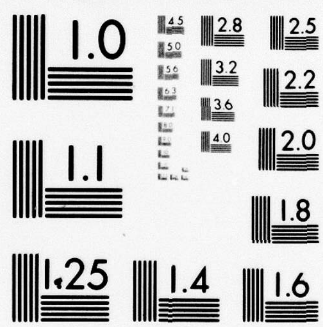

MICROCOPY RESOLUTION TEST CHART

REPORT NO. FPRC-9M1

A061383

MERADCOM/OSU HYDRAULIC SYSTEM RELIABILITY PROGRAM

Prepared by

FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA

MAY 1979

ANNUAL REPORT

May 1978 – May 1979

Approved for Public Release: Distribution Unlimited

Prepared for

U.S. ARMY MOBILITY EQUIPMENT RESEARCH
AND DEVELOPMENT COMMAND

Ft. Belvoir, Virginia 22060

Contract No. DAAK70-77-C-0097

14 OSU-
REPORT NO. FPRC-9M1

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15 Contract No. DAAK70-77-C-0097

409 263

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Fluid Power Research Center, Division of Engineering, Technology & Architecture, Oklahoma State University, Stillwater, OK 74074	**2a. REPORT SECURITY CLASSIFICATION** Unclassified **2b. GROUP**
3. REPORT TITLE MERADCOM/OSU HYDRAULIC SYSTEM RELIABILITY PROGRAM (Hydraulic Pumps, Cylinders, and Valves)	
4. DESCRIPTIVE NOTES (Type of report and inclusive dates) Final Report 1 June 1978 - 31 May 1979	
5. AUTHOR(S) (First name, middle initial, last name) Staff of the Fluid Power Research Center	
6. REPORT DATE May 1979	**7a. TOTAL NO. OF PAGES** 360 / **7b. NO. OF REFS** 20
8a. CONTRACT OR GRANT NO. DAAK70-77-C-0097 **b. PROJECT NO.** **c.** **d.**	**9a. ORIGINATOR'S REPORT NUMBER(S)** **9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)**
10. DISTRIBUTION STATEMENT Approved for Public Release: Distribution Unlimited	
11. SUPPLEMENTARY NOTES	**12. SPONSORING MILITARY ACTIVITY** Mobility Equipment Research & Development Command

13. ABSTRACT

The purpose of the Oklahoma State University/U.S. Army Mobility Equipment Research and Development Command Program is to provide the military with tools for the scientific appraisal of fluid power systems. Section I presents a detailed account of the project activities concerning hydraulic pump specifications. Included are an OSU proposed revision of Army pump specification MIL-P-52675 and two proposed test procedures for which no industrially recognized standards exist at present. Also included is a discussion and justification of the proposed specification and test procedures. Section II presents the details of the project activities concerning hydraulic valve specifications. Tests have been conducted based on Army specification MIL-V-52688 and the test results are discussed. OSU proposed revision of MIL-V-52688 is presented including additional test procedures that are considered necessary for Army valve specification. The project activities concerning hydraulic cylinder specifications are detailed in Section III. Included are an OSU proposed revision of Army cylinder specification MIL-L-52762. Also included is a summary of an industrial survey conducted on hydraulic cylinders and discussion and justification of the proposed specification and test procedures.

Unclassified

Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Hydraulic Pumps						
Positive Displacement Pumps						
Hydraulic Pump Durability						
Hydraulic Pump Contaminant Sensitivity						
Hydraulic Pump Performance Measurement						
Fatigue Failure Analysis						
Hydraulic Cylinders						
Hydraulic Valves						
Military Specifications						
Directional Control Valves						

MERADCOM/OSU HYDRAULIC SYSTEM RELIABILITY PROGRAM

SECTION I

HYDRAULIC PUMPS

PREPARED BY PERSONNEL OF

FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA

May 1979

FINAL REPORT

June 1978 – May 1979

PREPARED FOR

U.S. ARMY MOBILITY EQUIPMENT RESEARCH
AND DEVELOPMENT COMMAND

Fort Belvoir, Virginia 22060

U.S. ARMY MERADCOM

HYDRAULIC SYSTEM RELIABILITY PROGRAM

HYDRAULIC PUMPS

FINAL REPORT

PROJECT PERSONNEL

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PREFACE

This report presents a detailed account of the project activities concerning hydraulic pump specifications. Included are the OSU proposed revision of Army pump specification MIL-P-52675 and two proposed test procedures for which no industrially recognized standards exist at present. Also included is a discussion and justification of the proposed specification and test procedures.

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CHAPTER I

INTRODUCTION

The ultimate goal of the MERADCOM-OSU Hydraulic System Reliability Program is to exploit technology which has been developed and to apply this technology to induce more reliable mobile hydraulic systems on government-procured machinery. During previous MERADCOM-OSU programs, many test concepts and procedures for measuring the performance of hydraulic components were introduced to industry. The primary emphasis during this phase of the Hydraulic System Reliability Program was to obtain further industrial acceptance of those concepts and procedures in performance areas where no recognized standards exist. The availability of a full range of industrially-approved test procedures for pumps would end the need for the military to require "military-special" performance tests. Many manufacturers of quality pumps are reluctant to perform such special tests for one buyer, but could justify the expense of a full range of performance tests if such tests were universally acceptable and requested by customers other than the military.

Project personnel concluded that to complete the full set of desired test procedures for hydraulic pumps, a durability test procedure and a low temperature test procedure were needed to accompany the tests for hydraulic pumps already approved by the NFPA, SAE, and ISO. It was felt that the best approach to accomplishing the above objectives was to draft such test procedures and present them to the recognized standards organizations for further development and approval.

With the full range of tests for positive displacement hydraulic pumps at hand, a revised U.S. Army specification for hydraulic pumps was developed incorporating these test procedures. The purpose and use of the specification in the overall scheme of things, as seen by OSU personnel, is depicted in Fig. 1-1. The specification sets forth minimum performance requirements as measured using the set of test procedures now available. Upon meeting or surpassing these requirements, the pump is added to a Qualified Products List. The data obtained when performing the tests is forwarded to military personnel for use in selecting the "best" pump for a given application, as depicted in Fig. 1-2. The four major goals of the specification are:

1. Weed out any obviously deficient pumps.
2. Insure that U.S. Army personnel have adequate, accurate information to make the proper selection of the best candidate possible.
3. Insure that U.S. Army personnel have adequate, accurate pump performance information to design the most efficient hydraulic system.
4. Make use of test procedures acceptable to industry.

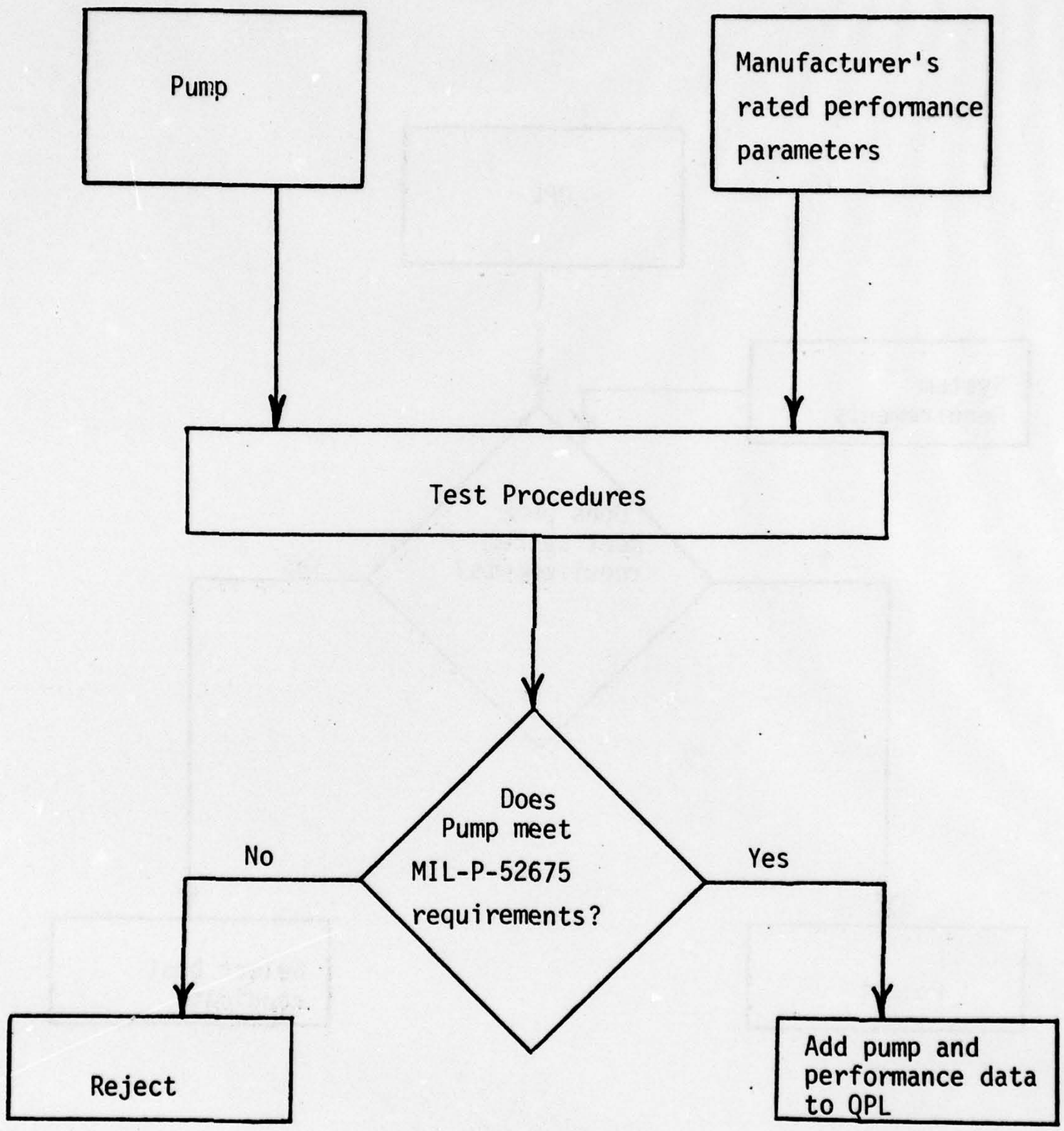

Fig. 1-1. Getting Pump on QPL.

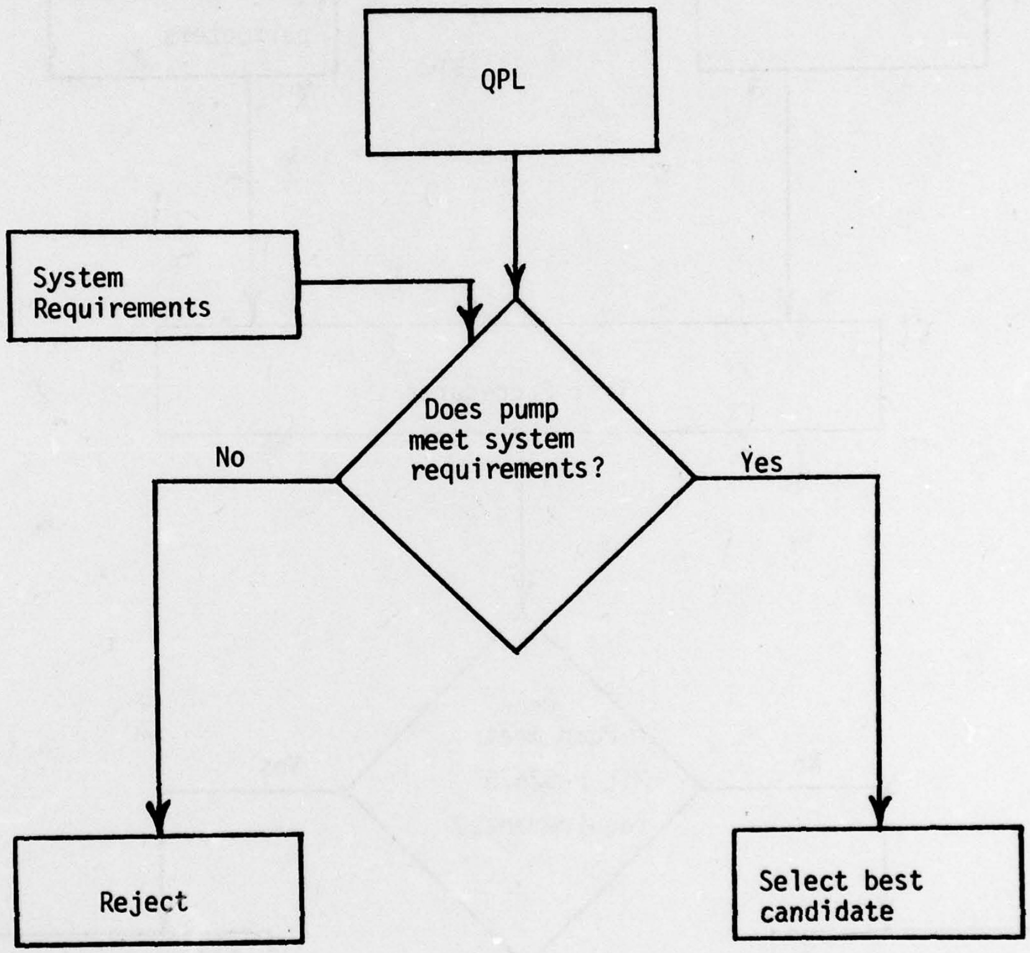

Fig. 1-2. Selection of Pump from QPL.

CHAPTER II

PROPOSED DOCUMENTS

The OSU proposed revision of military specification MIL-P-52675A is included in Appendix B. Major changes were made in the topics covered and the order of their presentation to increase clarity and to prevent repetition. Hopefully, the changes required in the document to suit military needs will be minimal.

The final version of the OSU-developed test procedure for determining pump durability, OSU-P-4-May 1979, is included in Appendix C. The final version of the OSU-developed test procedure for determining low temperature operation capability of a pump, OSU-P-6-May 1979, is included in Appendix D. It should be noted that several versions of these test procedures and the pump specification were developed and distributed during the project for discussion purposes. The documents in the appendices are the finalized versions and supersede all previous editions. The test procedures in Appendices C and D are the versions presented to the NFPA and SAE hydraulic pump working groups.

CHAPTER III

DISCUSSION OF THE REVISED PUMP SPECIFICATION

In the revised pump specification contained in Appendix B, a set of test procedures are called out for measuring pump performance. The results of these tests are used to determine whether or not the pump is of sufficient quality for use by MERADCOM. If the pump meets the minimum performance requirements, the data from the test are supplied to MERADCOM. In writing the specification, the following guidelines were observed:

1. Test procedures called out in the specification should be those most used and accepted in the fluid power industry.
2. The specification should be written such that the application of the specification and performance of the tests should be clear and unambiguous.
3. The test procedures called out in the specification and the order of the performance of the tests should be such that the most pump performance information may be obtained with the least possible effort by the party making the measurements.
4. The minimum performance requirements should be attainable by pumps presently on the market, yet high enough to insure that MERADCOM receives quality pumps and to serve as goals for pump designers.
5. The specification should remain as general as possible, to be applicable to any type or size positive displacement hydraulic pump.

A flow chart depicting the application of the developed specification is shown in Fig. 3-1. Each set of tests and minimum performance requirements shown will be discussed in detail in the following pages. During the discussions, it may be helpful to refer to Table 3-1, a comparison of positive displacement hydraulic pump test standards available.

The tests in the column under "OSU" in the table were developed by OSU in an earlier study [1]. The procedures and numbers in these tests were determined in meetings with representatives of fluid power component manufacturers and users. OSU project personnel feel the test procedures resulting from these meetings are quite valuable in the present project since they indicate what the industry agrees upon and uses in-house testing and measurement. The tests in the column under "MIL-P-52675A" are those contained in the current U.S. Army pump specification [2]. The column headed "MIL-P-52675B (First Draft)" contains summaries of tests outlined in a revision of MIL-P-52675A developed by MERADCOM personnel [3]. The final two columns in the table summarize two currently available positive displacement hydraulic pump test standards [5, 7].

DEFINITIONS

Strict definition of the terms used in fluid power component specifications or test procedures is important in creating documents which are not ambiguous. However, in many instances, this cannot be done due to the difficulty of making a strict definition suitable to all cases within the scope of a given specification or standard.

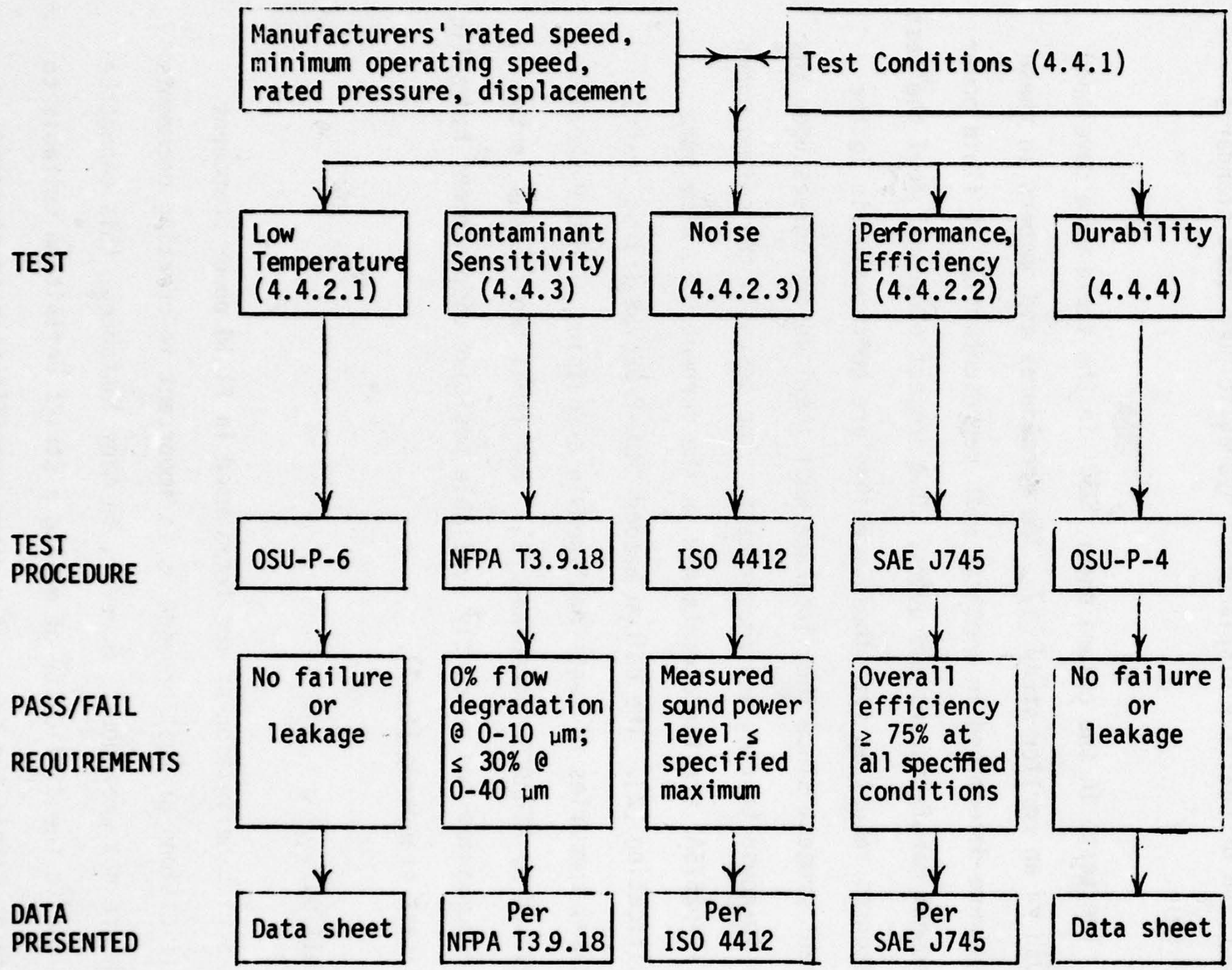

Fig. 3-1. Flow Chart—Application of the Specification.

TABLE 3-1. Comparison Table—Hydraulic Pump Test Standards.

TEST	PARAMETER	OSU	MIL-P-52675 A	MIL-P-52675 B (First Draft)	SAE J745c	ANSI/B93.27-1973 NFPA T3.9.17-1971
STRUCTURAL INTEGRITY	Fluid	MR	Mil-2104 Grade 10	Mil-2104 Grade 10	NONE	MR
	Inlet Temp	120°F	150°F	120°F		120°F
	Inlet Press	±1" Hg	Less than P_{atm}	1.2" Hg		MR or ±1.2" Hg
	Outlet Press	130% RP	3750 psi	115% MAOP		115% MAOP
	Speed	RS	RS	RS		RS
	Procedure	Run 60-70 sec.	Run for 60 sec.	Run 10 minutes		Run 10 minutes
DISPLACEMENT	Fluid	MR	Mil-2104 Grade 10	SEE EFFICIENCY	95-115 sus @ 120°F	MR
	Inlet Temp	MR	150°F		120°F	120°F, 180°F
	Inlet Press	±1" Hg	< -5" Hg		±1" Hg	MR
	Outlet Press	500 psi	2500 psi		100 psi	91.5 psi
	Speed	600 RPM, 50%-100%RS	600 RPM, RS, 115%RS		1000 RPM	1000 RPM
	Procedure	Record flow	Record flow @ 30min		Measure flow	Measure flow
LOW INLET PRESSURE	Fluid	MR	Mil-2104 Grade 10	NONE	95-115 sus @ 120°F	NONE
	Inlet Temp	MR	150°F		120°F	
	Inlet Press	-10" Hg	≤ -5" Hg		-5" Hg	
	Outlet Press	500 psi	2500 psi		100 psi-RP (3pts)	
	Speed	RS	600RPM, RS, 115%RS		Min to RS	
	Procedure	Record flow	Record flow @ 30min		Record torque, flow	
EFFICIENCY	Fluid	MR	Mil-2104 Grade 10	Mil-2104 Grade 10	95-115 sus @ 120°F 50-54 sus @ 180°F	MR
	Inlet Temp	MR	150°f	120°F, 180°F	120°F, 180°F	120°F, 180°F
	Inlet Press	MR	< P_{atm}	1.2" Hg	±1" Hg	MR
	Outlet Press	50%-100% RP (6pts)	1000-2500psi (7pts	800 psi-RP (8 pts)	100 psi-RP (3 pts)	91.5psi-RP (5 pts)
	Speed	Min to RS (6 pts)	2500 RPM	Min to RS (4 pts)	Min to RS (? pts)	Min to RS (4 pts)
	Procedure	Record torque, flow	Record torque, flow	Record torque, flow	Record torque, flow	Record torque, flow

*Refer to document MR-Manufacturer's Recommended RS-Rated speed PR-Rated pressure Min-Minimum recommended speed

(Continued on following page)

TABLE 3-1. Comparison Table—Hydraulic Pump Test Standards (Cont.).

TEST	PARAMETER	OSU	MIL-P-52675 A	MIL-P-52675 B (First Draft)	SAE J745c	ANSI/B93.27-1973 NFPA T3.9.17-1971
DURABILITY	Fluid	MR	Mil-2104 Grade 10	Mil-2104 Grade 10		
	Inlet Temp	MR	50hr@220°F, 950hr @200°F	220°F		
	Inlet Press	±1" Hg	$\leq P_{atm}$	1.2" Hg		
	Outlet Press	Cycled 5-115% RP	Cycle 5sec @ Pmin 5sec @ RP	Cycle 200psi-115%RP	NONE	NONE
	Speed	80% - 100% RS	RS	80% - 100%RS		
	Procedure	500,000cy@60cpm then 50hr@115%RP	Run 1000 hrs	120 hr @60cpm. then 50hr@115%RP		
LOW SPEED OPERATION	Fluid	MR	Mil-2104 Grade 10			
	Inlet Temp	MR	150°F			
	Inlet Press	±1" Hg	$\leq$ -5" Hg	SEE EFFICIENCY	SEE EFFICIENCY	SEE EFFICIENCY
	Outlet Press	RP	2500 psi			
	Speed	600 RPM	600 RPM			
	Procedure	Record flow @1min	Record flow @30min			
HIGH SPEED OPERATION	Fluid	MR	Mil-2104 Grade 10			
	Inlet Temp	MR	150°F			
	Inlet Press	±1" Hg	$\leq$ -5" Hg	NONE	NONE	NONE
	Outlet Press	RP	2500 psi			
	Speed	110% RS	115% RS			
	Procedure	Record flow@30min	Record flow@30min			
LOW TEMPERATURE	Fluid	MR	ConDN600,Emory3908	ConDN600,Emory3908		
	Inlet Temp	*	*	*		
	Inlet Press	$\geq$ MR	$\leq P_{atm}$	$\leq P_{atm}$		
	Outlet Press	*	*	*	NONE	NONE
	Speed	*	*	*		
	Procedure	Start-up @ -32°F	Start-up @ -50°F	Start-up @ -50°F		

*Refer to document MR-Manufacturer's recommended RS-Rated speed RP-Rated pressure Min-Minimum recommended speed

The definitions shown in section 3 of the OSU revision of MIL-P-52675A were included to eliminate ambiguity and provide clarity in applying the test procedures called out. Most of the definitions were taken directly from Ref. [4] with slight modification.

In many cases, the phrase "for the purposes of this specification" has been added to the ANSI definition. This is in recognition of the fact that a pump manufacturer may vary the value of a characteristic, such as rated speed, dependent upon the tests to be run or the operating conditions the user specifies. "For the purposes of this specification" means the vendor should specify values of rated speed, rated pressure, etc., for which the pump will pass the minimum performance requirements in this specification while allowing the maximum possible operating range in terms of speed, pressure, etc.

The defining of "volumetric displacement" proved to be quite a problem. The definition is very important because efficiency calculations make use of this value. Reference [4] defines volumetric displacement as "the volume for one revolution or stroke." However, delivery per revolution is dependent upon operating conditions of the pump such as speed, inlet pressure, outlet pressure, temperature, and fluid, as well as the operating history of the pump prior to measurement. To keep the volumetric efficiency in a range less than or equal to 100%, volumetric displacement must be greater than or equal to the maximum delivery per revolution measured at any time for a given pump.

This problem was solved by defining a "manufacturer's volumetric displacement" and a "measured volumetric displacement," with the volumetric displacement defined as the greater of the two. Measured volumetric displacement is the largest value of delivery per revolution experienced during any of the tests called out in the pump specification. Manufacturer's volumetric displacement is the value specified by the pump manufacturer for the purposes of this specification. A lower than actual value will be taken care of by the measured volumetric displacement. A higher than actual value will result in lower calculated efficiency and possibly failure to achieve minimum performance requirements.

REVERSABILITY

Concerning section 4.3.3, MERADCOM should consider making optional the requirement that a pump be reversible. This will eliminate many excellent pumps, and reversibility is totally impractical for some types of pumps.

TEST CONDITIONS

Section 4.4.1. entitled "Test Conditions," was included in the specification to take care of conditions which were common to all called out tests at one time rather than repeating the test conditions as each procedure is called out. Some comments on each subsection follow.

4.4.1.1 Oil: This statement supersedes any made in the individual test procedures, such as SAE J745 or ANSI/B93.27, with the exception of the low temperature test.

4.4.1.2 Filtration: This statement insures that the tests will not be run in a very dirty system. The actual numbers used in this statement are important only in the durability test and will be discussed later in this respect. The contaminant level in a reasonably clean system should have negligible affect upon the results of other measurements called out.

4.4.1.3 Instrument Accuracy: There was no special reason for choosing the values in ANSI/B93.27 over those of SAE J745 or making up a set of values. The limits in ANSI/B93.27 are adequate and convenient. Note that the accuracy for pressures below (or near) atmoshperic should be ±0.5 in Hg rather than ±2% as shown in most copies of the standard.

4.4.1.5 Tolerance of Test Parameters: Similar to "Instrumentation Accuracy" above, values of tolerance of test parameters used in ANSI/B93.27 are adequate and convenient. Note that the tolerance for inlet pressure should be ±1 in Hg or ±1.2 in Hg rather than ±2% as shown in most copies of the standard. Also note that the tolerances set forth here apply only if they are set forth no where else.

4.4.1.6 Run-In: Note that the phrase "for a 2-hour period" has been removed to avoid any inconsistencies which would occur if, for example, a manufacturer recommended a 1½-hour run-in.

4.4.1.7 Aeration: "The inlet oil must be visually free..." means provisions must be made to allow viewing of the inlet oil throughout testing.

4.4.1.8 Leakage: Any evidence of leakage other than at the shaft seal constitutes pump failure. The quantity of 1 drop per 10 minutes was agreed upon by MERADCOM and OSU personnel as allowable. The quantity is meant to be interpreted in the following manner: Observe the pump while running at rated speed, rated pressure for 10 minutes. If more than one drop falls, the pump has failed.

LOW TEMPERATURE TEST

The low temperature test is discussed later in this report where OSU-P-6, "Method for Establishing the Low Temperature Operation Ability of a Fixed Displacement, Fluid Power Pump," is examined. Note that the sequence of the tests called out in section 4.4.2 is important since the performance and efficiency test which follows will make evident any damage to the pump during the low temperature test.

PERFORMANCE AND EFFICIENCY

The procedures for measurement of performance and efficiency set forth in SAE standard J745 are used in this specification. ANSI/B93.27 contains essentially the same procedure for efficiency measurement, but was not used because it does not require performance measurement with low inlet pressure. The efficiency of the pump at such low inlet pressures is an indication of the susceptibility of the pump to cavitation. If for some reason the use of ANSI/B93.27 in place of

SAE J745 in the specification is desired, a low inlet pressure test could be written and called out separately from the performance and efficiency test.

Another reason, somewhat trivial, for not using ANSI/B93.27 is the structural integrity test incorporated in the procedure. Since we recommend a structural integrity test on every pump produced (see clause 7.4.2 of the proposed specification), the test need not be repeated on the pump used for the performance and efficiency test.

The value of 75% for minimum overall efficiency is arbitrary in that we have no good arguments why it should be 75% rather than 80% or 75% rather than 70%. The value of 75% just seemed to be a good value based upon pump performance data we have seen and experienced. Obviously, the needed overall efficiency throughout the operating range will vary widely with the application for which the pump is being considered. During discussions of the proposed specification, the point was often made that requiring such a "high" value of overall efficiency might work as a disadvantage if the pumps satisfying the requirement are very expensive. However, any pump will pass the 75% overall efficiency requirement if the operating ranges (speed and outlet pressure) are sufficiently small. For example, a pump with manufacturer specified rated pressure of 3000 psi, rated speed of 300 rpm, and minimum operating speed of 200 rpm may yield a minimum overall efficiency of perhaps 20% somewhere in this operating range. If the manufacturer of this pump specifies a smaller operating range, say, rated pressure of 200 psi, rated speed of 1500 rpm, minimum operating

speed of 800 rpm, the minimum overall efficiency as determined using this specification could be "raised" to 80%. In this way, the operating range and minimum overall efficiency go hand-in-hand. The range of operation specified by the pump manufacturer will be limited to the range in which the pump performs well rather than being allowed to vary from zero up to the maximum values of outlet pressure and speed which the pump can mechanically endure.

Limiting the operating range in this way would have prevented a problem experienced by MERADCOM personnel on a hydraulic steering system. When the drive motor in this system was idling, loading of the pump resulted in excessive slip flow (hence, very low efficiency) which lead to excessive heating of the pump and no steering power. If the operating range over which the pump performed with a minimum overall efficiency of 75% had been known, the problem could have been spotted and rectified in the design stage.

PUMP NOISE GENERATION

Although only a Draft International Standard at present, ISO/DIS 4412 appears well on the way to approval. OSU personnel believe it is an excellent document and the only complete pump noise measurement procedure available. The specified test conditions, rated speed, rated pressure, and inlet temperature of 180°F are the conditions within the manufacturer's specified operating range at which most pumps will generate the greatest amount of noise.

A single value could be used for the maximum allowable overall sound power level, but a curve allowing greater sound power levels with increasing horsepower was used instead. (In general, as the horsepower of the pump increases, the drive and other components in the hydraulic system emit more noise.) A plot of sound power level (in dBA re 10^{-12} watt) versus horsepower for a number of pumps is shown in Fig. 3-2. The pumps whose sound powers are plotted were of various types and sizes. The upper line across the plot is the maximum allowable sound power level set forth in the specification. The slope of this line, 3 dB per doubling of horsepower, was chosen based on experience and fit to the data shown. (The pumps for which we have data in the 5 to 10 hp range were known to be "quiet" pumps.) With this value of slope, the maximum allowable noise line was positioned vertically such that pumps with noise levels more than 5 dB greater than average for that hp were rejected. Note that the resulting limit set forth in the specification, $W = 10 \log_{10} (hp) + 74$, was achieved by more than 90% of pumps for which OSU has noise data.

CONTAMINANT SENSITIVITY

The test called out in the specification, NFPA/T3.9.18—Method for Establishing the Flow Degradation of Hydraulic Fluid Power Pumps When Exposed to Particulate Contaminant, sets forth a procedure to obtain pump flow degradation values after 0-5, 0-10, 0-20, ..., 0-80 micrometre injections of classified AC Fine Test Dust. These tests are usually performed at operating conditions of rated speed, rated pressure, and oil temperature of 180°F; hence, these conditions are given in the specification.

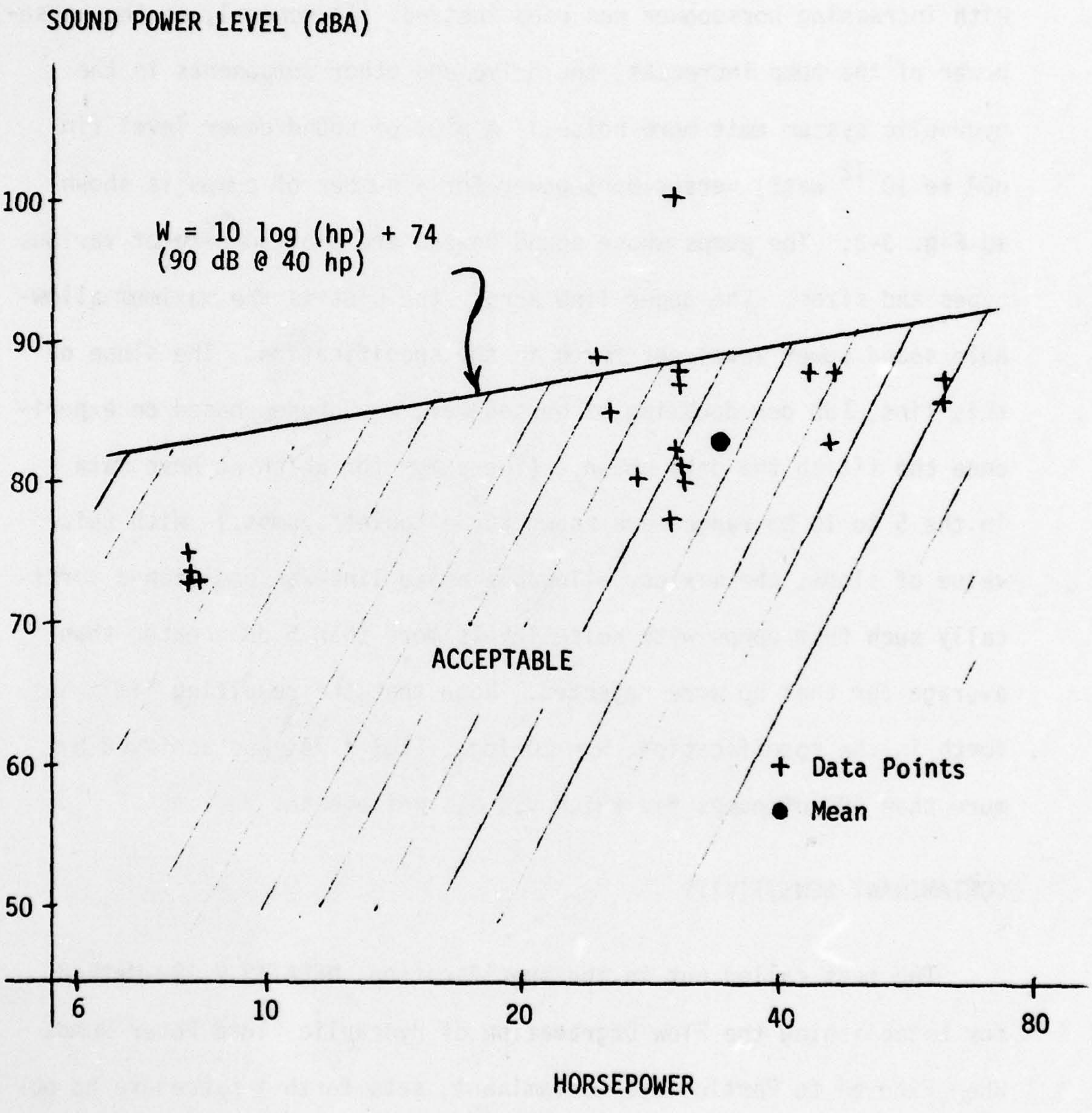

Fig. 3-2. Sound Power Level Vs. Horsepower.

The limits for flow degradation can be explained most easily using the Omega Pump Rating System of Fig. 3-3. (See Ref. [10] for more information concerning the Omega Rating System.) The Omega Rating is a single number which indicates the contaminant sensitivity of the pump. It has been found that for the filtration available and the environments in which U.S. Army hydraulic systems operate, a pump should have an Omega Rating of 2.2 or less to give satisfactory performance for 1000 hours. Of course, this number is only approximate as contaminant wear is dependent upon a number of factors which vary from system to system. The mean Omega Rating of pumps tested at OSU is approximately 3.0; hence, a value of 2.2 is well within the capabilities of most pump manufacturers.

To achieve an Omega Rating of 2.2 or less, the flow degradation after the 0-5 μm and the 0-10 μm injections must be less than or equal to zero. Drawing a straight line from the 2 "no-degradations" points on the Y axis through the Omega = 2.2 point establishes the X-axis intercept shown. A line through the X-axis intercept and intercept value of 4.1 on the horizontal "Intercept Value" axis yields maximum flow degradations of about 1% after the 0-20 μm injection, about 8% after the 0-30 μm injection, and about 30% after the 0-40 μm injection. In most cases, however, if the 0-40 μm criterion is met, the 0-20 and 0-30 will satisfy the requirements also; hence, only the 0-40 requirement appears in the specification.

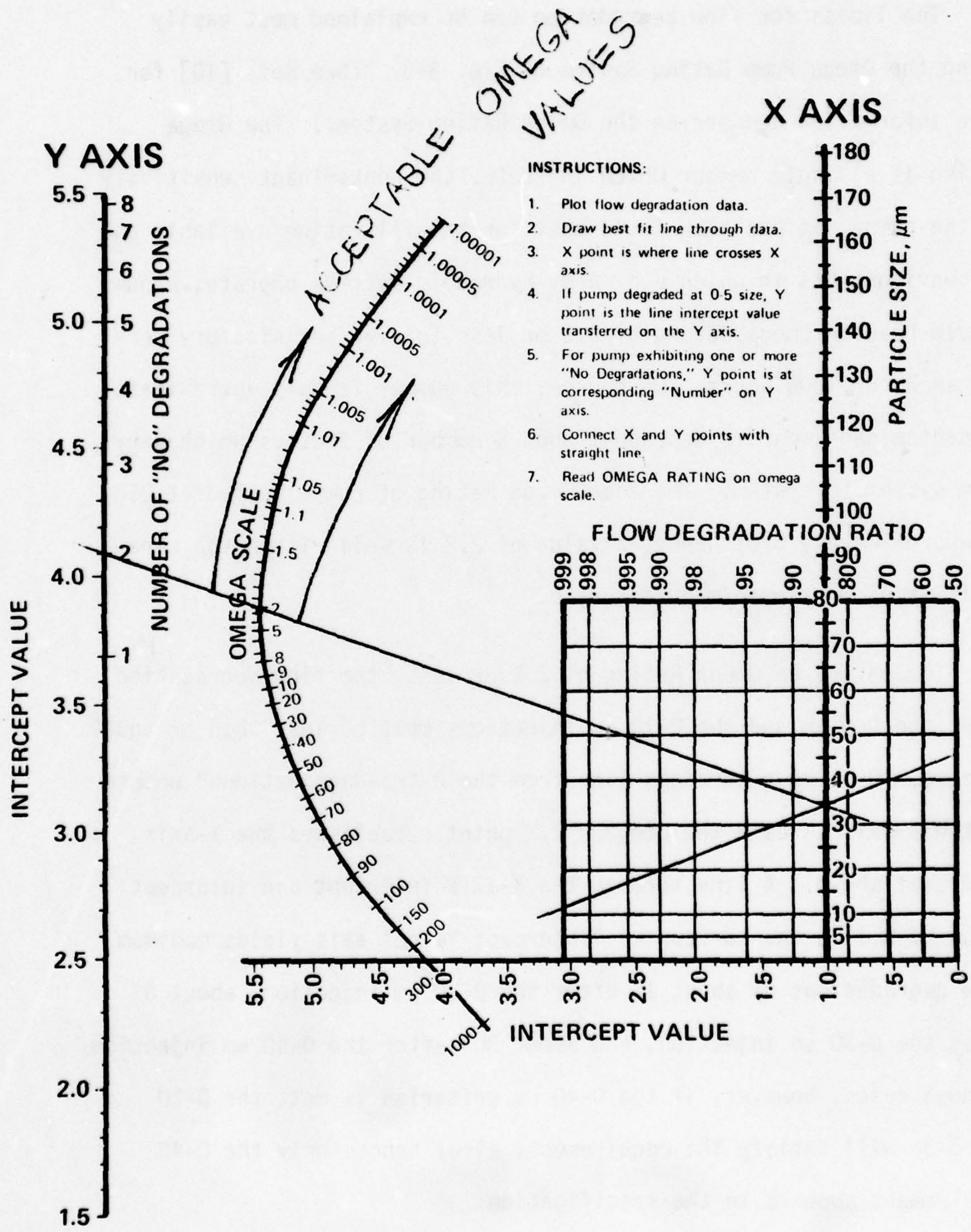

Fig. 3-3. Omega Pump Rating Chart.

DURABILITY TEST

The durability test is discussed later in this report where OSU-P-4, "Method for Establishing the Durability of Positive Displacement Fluid Power Pumps," is examined.

PRODUCTION TESTS

Production testing on every pump produced is necessary for quality control purposes. All manufacturers do some type of production testing, but perhaps not at the speed, pressure, and for the length of time in section 7.4.2. By using a pressure above rated pressure and a slightly longer than usual test time length, a structural integrity test is performed on each pump.

The use of five minutes for length of the test has been questioned. Five minutes is a compromise value; less than one minute of testing might not bring out defects in the pump, but increasing the test time increases the cost per pump. While 10 minutes or 15 minutes might be preferable, five minutes per pump is all that can be reasonably expected of pump manufacturers.

CHAPTER IV

DISCUSSION OF THE DURABILITY TEST

The section in the specification calling out the durability test and the durability test procedure, OSU-P-4-May 1979, will both be discussed here. Before examining the procedure and specifics, a general discussion of cyclic endurance testing will be presented.

At a given endurance test condition, the "life" of a component for a given mode of failure is dependent on the cumulative failure distribution for that component population in that failure mode. As shown in Fig. E-5, the location, "minimum life," and shape of the cumulative failure distribution is a function of the test conditions or effective "strain" per cycle.

The behavior illustrated in Fig. E-5 is typical. For instance, Fig. E-4 shows a failure distribution for an aluminum pump population.

As discussed in Appendix E, if only one pump from the population (shown in Fig. E-4) is tested, there is a 10% chance that it will survive beyond 4,400,000 cycles. Following this line of reasoning, if testing is only conducted to one million cycles on each pump, it becomes necessary to test more than one unit to gain confidence that a set of samples is from the population shown in Fig. E-4.

Since we generally desire some knowledge about the "minimum" life of a population, it would be best to test such that we could demonstrate that the sample population meets the minimum life require-

ments. To do this with minimum risk, we would need the failure distribution of the population. Since the failure distribution is not always available, or allowing it to be a variable greatly complicates acceptance, we can take the alternative of assuming the shape of the distribution, "building in" a degree of conservatism, and establishing a procedure which provides knowledge about product reliability with reasonable confidence or minimum risk.

Appendix E provides the information for establishing Table E-2 and Fig. E-7. What these show is that if one desires a minimum life of 500,000 cycles for the distribution as is shown and also desires a consumer's risk of 10% or less, then if only one unit is tested, it must survive 3,125,000 cycles. Likewise, if five units are tested, they must each survive 625,000 cycles.

It would be premature to write specification that fully implements the implications in Appendix E. An intermediate position is to recognize the requirement outlined in Appendix E and establish a reasonable procedure which can be "updated" after more effort is directed toward developing more exacting acceptance procedures.

Recognizing the need for verified reliability, the reluctance to conduct extensive amounts of testing, the limits of our knowledge about the influence of "accelerated" loads above 1.2 rated condition, and the lack of adequate failure distributions for all materials, the following life ratios and lives are recommended for verification testing when the minimum desired test life is in the vicinity of 400,000 cycles.

TABLE 4-1. Minimum Number of Cycles to Failure for "N" Pumps with a Desired Life of 400,000 Cycles.

N—Number of Pumps Tested	Minimum Life for Each of the "N" Pumps Tested	Life Ratio
2	1,500,000	3.75
3	900,000	2.25
4	650,000	1.63
5	500,000	1.25

The option of testing only one unit is eliminated because for certain materials (See Fig. 4-1, *taken from* Ref. [11].), one unit would have to survive over 20,000,000 cycles for the desired minimum of 400,000 cycles. However, the difference in failure distributions for different materials becomes smaller as the distributions approach the "minimum" life. Therefore, for two or more units, it seems reasonable to use the life ratios in Table 4-1 and subsequently adjust reliability calculations based on available information regarding component materials and their associated failure distributions.

Generally, verification testing should be conducted for more than 100,000 cycles because short life tests can lead to erroneous decisions about life in the field (or long life at reduced loads).

The actual life numbers used in the specification will have to be decided by MERADCOM based on their reliability needs. For instance, if the desired life in the field is 10,000 hours and the acceleration

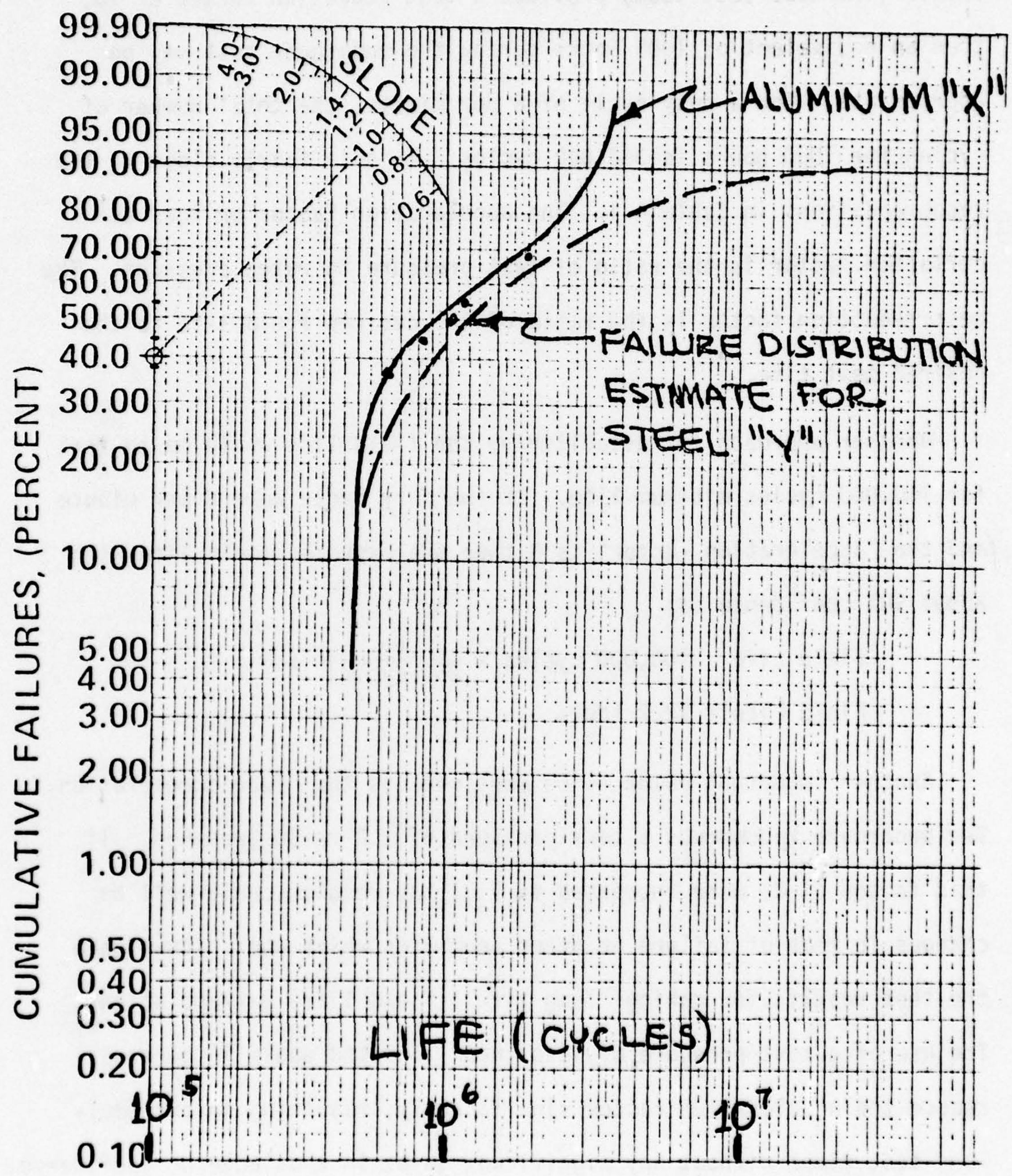

Fig. 4-1. Graphical Presentation of Failure Distributions for Aluminum "X" and Steel "Y" (from Ref. [11]).

factor (increase test load) provides a test reduction factor of 10, then an equivalent of 1000 hours during the endurance test may be appropriate. For a duty cycle of 6 per minute, the total number of cycles for 1000 hours is 360,000 cycles. This is fairly close to the recommendations in Table 4-1. The acceleration factor or load accerlation factor is the ratio of test pressure to rated pressure. The test reduction factor is the ratio of the desired field life to the desired test time.

Another approach for establishing the field life would be to test for 100,000 cycles minimum life. If the duty cycle were 6 per minute and the "acceleration factor" is 8 then the expected field life with about 90% confidence is:

$$\text{Field Life} = 100{,}000 \text{ cycles} \times \frac{\text{hr}}{360 \text{ cycles}} \times 8$$

$$\text{Field Life} = 2222 \text{ hours}$$

Another important point to be mentioned is the "load acceleration." The procedure recommends a test pressure of 1.2 rated pressure. If this factor (1.2) were increased to 1.3, the results could well be changes in mode of failure or other anomalies which would invalidate the test results for making "long life" (field life) extrapolations. The use of a test pressure of less than 1.1 rated would in general reduce the acceleration factor significantly, resulting in extremely long test times without any significant gains in knowledge or confidence.

The use of a 1.2 rated pressure for testing is fairly common in the fluid power industry. If a procedure is desired which used only one test pressure, then 1.2 times rated is a very reasonable figure.

In the hope of clarifying these matters, two examples will be presented.

Example 1

Suppose MERADCOM personnel need a pump for an application whose duty cycle may be modeled by a six cycle per minute square wave outlet pressure, varying from near atmospheric to rated pressure during each cycle. Calculate the number of cycles to failure for "N" pumps from the production lot to be tested, using OSU-P-6-May 1979 if a 10% reliability of operating 10,000 hours is desired.

Solution: Assume the increase in pressure from 1.0 to 1.2 will decrease the life expectation by a factor of 9.0.

$$(10{,}000 \text{ hours}) \times (360 \text{ cycles/hour}) \div 9 = 400{,}000 \text{ cycles}$$

The failure distribution will be assumed equal to that shown in Fig. 4-1 for aluminum. From this distribution, the life ratio for "N" pumps may be determined, using the cumulative failure percentages found in Appendix E. The results are shown in Table 4-2. (These are the values suggested by OSU for MERADCOM to use in the durability test requirements.)

TABLE 4-2. Example 1 Results.

N—Number of Pumps Tested	F(x) (See Appendix E)	Life Ratio (From Fig. 4-1)	Minimum Life for Each "N" Pump
1	90.0%	10.00-up	4,000,000-up
2	68.4	3.75	1,500,000
3	53.6	2.25	900,000
4	43.8	1.63	650,000
5	36.9	1.25	500,000

Example 2

Rework example 1 assuming only 75% reliability is necessary.

Solution: The cumulative failure F(x) associated with "N" pumps is given by

$$F(x) = 1 - (0.25)^{1/N}$$

The values of F(x) for N = 1, 2, ..., 5 are shown in Table 4-3. Defining the life ratio as L_i/L_{50}, the distribution from Fig. 4-1 gives the ratios shown in the table.

TABLE 4-3. Example 2 Results.

N	F(x)	Life Ratio L_i/L_{50}	Minimum Life
2	50.0%	2.10	840,000
3	37.0	1.45	580,000
4	29.3	1.25	500,000
5	24.2	1.15	460,000

Although these results may at first appear appealing for this example, there is one chance in four of being wrong.

The defining of pump failure presents a difficult problem in the durability test procedure. If catastrophic failure does not occur, then the point at which the pump fails to provide adequate performance may be used as the moment of failure. However, removing the pump from the endurance test system and measuring overall efficiency throughout the pump's operating range periodically during cyclic testing would prove very inconvenient and expensive. By requiring only volumetric efficiency to be measured and at only one set of operating conditions, the point of failure problem is eliminated. Provisions for the

measurement of flow at steady outlet pressure can easily be incorporated into the cyclic endurance test system.

The maximum flow degradation at which pump performance is still adequate varies with application and priorities. For this reason, the value is not set in the durability test procedure but is left to be specified by the test sponsor. OSU personnel recommend a value of 75% volumetric efficiency at rated speed and rated pressure.

The test conditions in the durability test procedure are quite similar to those used for the specification and were discussed earlier. The filtration requirement will be discussed here, since it is very important to the durability test.

The durability test should be a measurement of a pump's resistance to fatigue failure and not contaminant wear failure. The cleanliness level stated in the test procedure would provide at least 1000 hours of constant pressure running at above 70% volumetric efficiency, assuming the pump met the contaminant sensitivity test requirements. The pressure cycling reduces this to the 500-600 hour range. Hence, it is desirable to complete the endurance testing on a pump quickly not only for convenience but also to insure that contaminant wear is not a significant factor. Cycling at 60 cycles per minute rather than at six cycles per minute is quite advantageous in this respect.

The tolerance on the speed during cyclic testing is relatively large. The on/off pressure cycling makes steady control of the speed quite difficult unless the pump and drive system have very large rotational inertia.

OSU personnel made use of the waveform described in SAE J343, since this waveform is well suited to cyclic testing. Note that no limits are placed upon the rate of pressure increase or the rate of pressure decrease; hence, they could approach infinite values. However, values above 150,000 psi/sec for the rate of pressure increase are somewhat difficult to obtain unless the outlet system and load valve are designed specifically for that purpose. Concerning the "fall" portion of the waveform, the rate of pressure decrease does not appear to have significant influence on the total strain per cycle, since the significant strain has occurred by the end of the time at pressure. Some concern has been voiced regarding the possibility of the rapid pressure decreases leading to cavitation. However, there is no known documentation to indicate that, if the inlet fluid is free of air as specified in the test procedure, there would be any subsequent cavitation problem at the high pressure outlet even at low pressures.

CHAPTER V

DISCUSSION OF THE LOW TEMPERATURE TEST

The section in the specification calling out the low temperature test and the low temperature test procedure, OSU-P-6-May 1979, will both be discussed here. A step-by-step discussion of the important aspects of the procedure follows.

4.3 Oil: The type of fluid to be used appears to be the most important aspect of the test, and the area of most interest to those in industry with whom the low temperature test was discussed. It is suggested that more research be done by MERADCOM to evaluate the degree of variation in properties between the different low temperature fluids on the market and differences from one container to the next of what is supposedly the same oil. Experience in the area of low temperature fluids seems to indicate wide variation.

4.4 System Volume: This restriction is necessary to control the rate at which the system heats up. The value used here was agreed upon by representatives of the fluid power industry as previously discussed.

5.4, 5.5, 5.6 The temperatures used here are indications of what we feel are the worst conditions under which the pump might be expected to perform. Note that the pump is not operated at cold soak temperature of -65^{o}F; this simply a test of the pumps ability to operate after being exposed to this temperature.

The test schedule of section 5.6 is a simulation of start-up under worst-case conditions. The procedure here is very similar to that in the original OSU-P-6 and that in MIL-P-52675A. The original OSU-P-6 test schedule was approved by representatives of the fluid power industry as previously discussed. In recent discussions, it has been pointed out that very little low temperature testing is being done in industry upon single components. The more common approach is the observation of the effects of low temperatures on complete hydraulic systems. In any case, the low temperature test would appear to be the major obstacle in persuading pump manufacturers to perform self-certification. For this reason, making this an optional test might be in the best interest of MERADCOM.

5.7 Further defining of "malfunction" would be pointless due to the wide variety of unexpected problems which might be encountered at low temperatures. Note that the same pump used in the low temperature test is later used for efficiency and performance tests. Any damage due to exposure and operation at low temperature should show up at that point. Therefore, for the purposes of the MERADCOM specification, detailed performance measurements upon the pump are not needed within the low temperature test procedure.

CHAPTER VI

PRESENTATION TO INDUSTRY AND ADDITIONAL COMMENTS

Throughout the project, OSU personnel attended meetings of the NFPA and SAE pump working groups to keep them informed of our work and to obtain their views on these docuements. The two test procedures are currently under consideration by both organizations. Final versions of the test procedures, as contained in the appendices, and applicable portions of this report will be presented to the appropriate personnel in each organization to aid in their review of the two documents. We believe that through close work between MERADCOM personnel and these representatives of the fluid power industry, excellent progress will be made.

We are deeply indebted to our friends in the industry who helped in this project by offering their criticism and comments. Where possible, these suggestions have been incorporated into the documents.

APPENDIX A

REFERENCES

1. OSU Hydraulic Pump Test Procedures, P-1 through P-7, U.S. Army MERADCOM Hydraulic Specification Program.

2. Mil-L-52675A (22 August 1973) Military Specification - Pumps, Hydraulic, Petroleum Base Oil, Fixed Displacement

3. Mil-L-52675B (28 April 1978) Proposed Revision of Mil-L-52675A, Authored by MERADCOM Personnel.

4. ANSI/B93.2 - American National Standard Glossary of Terms for Fluid Power.

5. ANSI/B93.27 - American National Standard Method of Testing and Presenting Basic Performance Data for Positive Displacement Hydraulic Fluid Power Pumps and Motors.

6. NFPA/T3.9.18 - Method of Establishing the Flow Degradation of Hydraulic Fluid Power Pumps When Exposed to Particulate Contaminant.

7. SAE J745 - Hydraulic Power Pump Performance Characteristics Measurement and Presentation.

8. SAE J343 - Tests and Procedures for SAE 100R Series Hydraulic Hoses and Hose Assemblies.

9. ISO/DIS 4412 - Hydraulic Fluid Power - Pumps - Test Code for Determination of Airborne Noise Levels.

10. Bensch, L. E., and E. C. Fitch, "Specifying Fluid Cleanliness and Filtration Requirements Using Component Contaminant Sensitivity Data," The BFPR Journal, 1979.

11. Maroney, G. E., "Production Quality Control Fatigue Testing Decisions Based on Material Properties," The BFPR Journal, 1980.

APPENDIX B

DRAFT

MILITARY SPECIFICATION

PUMPS, HYDRAULIC, PETROLEUM BASE OIL, FIXED DISPLACEMENT

1. SCOPE: This specification covers fixed displacement hydraulic pumps for use on stationary and mobile type equipment. The pumps covered by this specification are intended for production use, as spares, and as replacements in hydraulic systems or stationary and mobile-type military equipment.

2. APPLICABLE DOCUMENTS

2.1 The following documents of the issues in effect on date of invitation for bids or request for proposal form a part of this specification to the extent specified herein:

SPECIFICATION

Federal

PPP-T-60	-Tape, packaging, waterproof.
PPP-B-601	-Boxes, wood, cleated-plywood.
PPP-B-636	-Boxes, shipping, fiberboard.
QQ-S-781	-Strapping, steel and seals.

Military

MIL-B-121	-Barrier material, greaseproofed, waterproofed, flexible.
MIL-P-514	-Plates, identification, instruction and marking, blank.
MIL-L-2104	-Lubricating oil, internal combustion engine, tactical service
MIL-P-5514	-Gland design, packings, hydraulic general requirements for.
MIL-H-5606	-Hydraulic fluid, petroleum base; aircraft and ordance.

MIL-H-6083 -Hydraulic fluid, petroleum base; preservation and testing.

STANDARDS

MIL-STD-105 -Sampling procedures and tables for inspection by attributes.

MIL-STD-129 -Marking for shipment and storage.

MIL-STD-130 -Identification marking of U.S. military property.

MIL-STD-889 -Dissimilar metals.

MIL-STD-1188 -Commercial packaging of supplies and equipment.

(Copies of specifications and standards required by the suppliers in connection with specific procurement function should be obtained from the procuring activity or as directed by the contracting officer).

2.2 Other Publications. The following documents form a part of this specification to the extent specified herein. Unless otherwise indicated, the issue in effect ondate of invitation for bids or request for proposal shall apply;

NATIONAL BUREAU OF STANDARDS

Handbook H28 - Screw-Thread Standards for Federal Service.

(Application for copies should be addressed to the Superintendent of Documents, Government Printing Office, Washington, D.C. 20402.)

AMERICAN NATIONAL STANDARD INSTITUTE

ANSI/B93.2 - American National Standard glossary of terms for fluid power.

ANSI/B93.27 - American National Standard method of testing and presenting basic performance data for positive displacement hydraulic fluid power pumps and motors.

ANSI/B93.35 - American National Standard groove dimensions for fluid power exclusion devices (inch series).

(Application for copies should be addressed to the American National Standards Institute, 1430 Broadway, New York, N.Y. 10018).

NATIONAL FLUID POWER ASSOCIATION, INC.

NFPA/T3.9.18 - Method of establishing the flow degradation of hydraulic fluid power pumps when exposed to particulate contaminant.

(Application for copies should be addressed to the National Fluid Power Association, P.O. Box 49, Thiensville, WI 53092).

SOCIETY OF AUTOMOTIVE ENGINEERS

SAE Handbook

SAE Standard

SAE-J514 - Hydraulic tube fittings.

SAE-J518 - Hydraulic flanged tube, pipe and hose connections, 4 bolt split flange type.

SAE-J744 - Hydraulic power pump and motor mounting flange and shaft.

SAE-J745 - Hydraulic power pump performance characteristics measurement and presentation.

SAE-J343 - Tests and Procedure for SAE 100R Series Hydraulic Hose and Hose Assemblies.

(Application for copies should be addressed to the Society of Automotive Engineers, 400 Commonwealth Drive, Warrendale, PA 15096).

INTERNATIONAL STANDARDS ORGANIZATION (ISO)

ISO/R 1800 - Dimensions of elastomeric toroidal sealing rings (inch series - class 2 tolerance).

ISO/DIS 4412 - Hydraulic fluid power - pumps - test code for determination of airborne noise levels.

(Application for copies should be addressed to the American National Standard Institute, 1430 Broadway, New York, N.Y. 10018).

OKLAHOMA STATE UNIVERSITY

OSU-P-4-May 1979 - Method for establishing the durability of fixed displacement, fluid power pump.

OSU-P-6-May 1979 - Method for establishing the low temperature operation ability of a fixed displacement, fluid power pump.

(Application for copies should be addressed to the Fluid Power Research Center, Oklahoma State University, Stillwater, OK 74074).

3. DEFINITIONS

3.1 Hydraulic pump: A device which converts mechanical force and motion into hydraulic fluid power.

3.2 Fixed displacement hydraulic pump: A hydraulic pump in which the displacement per cycle cannot be varied.

3.3 Rated pressure: The qualified operating pressure which is recommended for a component or a system by the manufacturer for the purposes of this specification.

3.4 Rated speed: The maximum qualified operating speed which is recommended for a component or a system by the manufacturer for the purposes of this specification.

3.5 Minimum operating speed: The minimum qualified operating speed which is recommended for a component or a system by the manufacturer for the purposes of this specification.

3.6 Volumetric displacement: The volume for one revolution or stroke. For the purposes of this specification, volumetric displacement is equal to manufacturer's volumetric displacement or measured volumetric displacement, whichever is greater in magnitude.

3.7 Manufacturer's volumetric displacement: The maximum volume delivery of the pump per revolution, as specified by the manufacturer.

3.8 Measured volumetric displacement: The maximum volume delivery of the pump per revolution measured during any of the tests in this specification.

3.9 Flow at rated speed: The product of volumetric displacement and rated speed.

3.10 Direction of shaft rotation: Clockwise or counterclockwise as viewed from the shaft end of the pump.

3.11 For definitions of other fluid power terms used, see ANSI/B93.2.

4. PUMP REQUIREMENTS

4.1 Identification markings: The pump is identified, with markings which conform to MIL-STD-130, on identification plates which conform to MIL-P-514, (type III, composition A, class 1) with the following information:

PUMP, HYDRAULIC, OIL, FIXED DISPLACEMENT

RATED PRESSURE________________bar (psi)
RATED SPEED________________rpm
VOLUMETRIC DISPLACEMENT________________ℓ(in^3)
MINIMUM OPERATING SPEED________________rpm
FLOW @ RATED SPEED________________ℓ/min (gpm)
MFR. NAME________________
MFR. PART NO.________________
MFR. SERIAL NO.________________
GOVERNMENT PART NO.________________

4.2 Materials: Materials are new and conform to this specification.

4.2.1 Metals: All metals are compatible with the oil, temperature, function, service and storage conditions specified herein. Unless protected against electrolytic corrosion, dissimilar metals as defined in MIL-STD-889 are not used in intimate contact with each other.

4.2.2 Material compatibility: All materials of the pump are compatible with lubrication oils conforming to MIL-L-2104, Grade 10; CONOCO DN-600, Type I; EMORY 3908; and hydraulic fluids conforming to MIL-H-5606, MIL-H-6083, MIL-H- 83282, and MIL-H-46170.

4.3 Structural design:

4.3.1 Mounting flange and drive shaft: The pump conforms to mounting flange and drive shaft SAE Standard J744.

4.3.2 Ports: The pump ports, 3/4 inch and smaller, able to accept straight thread connectors must conform to SAE Standard J514. Port sizes larger than 3/4 inch must be 4 bolt split flange type connectors conforming to SAE J518.

4.3.2.1 Outlet ports must be large enough to limit outlet velocity to a maximum 6 m/s (20 ft/s) at rated speed.

4.3.2.2 Suction ports must be large enough to limit inlet velocity to a maximum 1.8 m/s (6 ft/s) at rated speed.

4.3.3 Direction of rotation: The pump provides clockwise or counterclockwise rotation. Rotation can be reversed by dismantling and repositioning internal components. Change of direction of rotation can be accomplished without addition, removal or replacement of any component. Direction of rotation is to be permanently marked on a reversible metal tag attached by screws to the pump mounting flange.

4.3.4 Seals and packing: The construction of the pump toroidal sealing rings (o-ring) conform dimensionally to ISO/R 1800. The press type double lip exclusion device groove dimensions conform to ANSI/B93-35. The installation and use of these sealing rings and exclusion devices conform to MIL-P-5514.

4.3.5 Workmanship: The pump assembly is clean and free from harmful extraneous material and all parts are the highest quality produce in keeping with good commercial practices.

4.4 Performance: Pumps from each production lot should be subjected to the following tests to ensure that the pumps meet minimum performance requirements and to obtain performance data to be provided to the procurer.

4.4.1 Test Conditions

4.4.1.1 Oil: All tests, except the low temperature test of section 4.4.2.1, will be conducted using lubricating oil MIL-L-2104, grade 10.

4.4.1.2 Filtration: Unless otherwise specified in the test procedure, provide a filter which will limit the total number of particles in the system fluid to 1000 particles per millilitre greater than 10 microns.

4.4.1.3 Instrumentation accuracy: Unless otherwise specified in the test procedure, select and maintain instrumentation so that measurements are accurate within the limits set forth in ANSI/B93.27.

4.4.1.4 Inlet pressure: Unless otherwise specified in the test procedure, maintain the inlet oil pressure at atmospheric pressure ±1 in Hg.

4.4.1.5 Tolerance of test parameters: Unless otherwise specified in the test procedure, maintain test parameters within the limits set forth in ANSI/B93.27.

4.4.1.6 Run-in: Before the tests are carried out, "run-in" the pump in accordance with manufacturer's recommendations.

4.4.1.7 Aeration: Minimize fluid aeration by taking precautions such as proper system design and by adequate removal of air from the system before testing. The inlet oil must be visually free of entrained air throughout testing.

4.4.1.8 Leakage: The pumps shall show no evidence of external leakage when tested as specified herein, except that shaft seal leakage of up to 1 drop per 10 minutes is allowable after subjecting a pump to the durability test of section 4.4.4.1.

4.4.2 Pump number one should be subjected ot the tests described in sections 4.4.2.1, 4.4.2.2, and 4.4.2.3, in that sequence.

4.4.2.1 Low temperature: The pump should be subjected to the low temperature test specified in OSU-P-6 May 1979. Use Conoco DN600, Type I, Emory 3908, or equivalent hydraulic oil.

4.4.2.2 Performance and efficiency: Pump performance and efficiency should be measured and reported in accordance with SAE J745 for the five values of outlet pressure at each of four speeds (pressure and speed test points are to be evenly distributed in the ranges pressure and speed ranges specified in SAE J745). If the volumetric displacement, as defined in 3.6 is greater than (SAE Volumetric Rating ÷ 1000), use the volumetric displacement in place of the SAE Volumetric Rating in efficiency calculations. Overall efficiency must be greater than or equal to 75% for all specified test points between and including minimum operating speed and rated speed.

4.4.2.3 Pump noise generation: Noise generated by the pump should be measured and reported in accordance with ISO/DIS 4412 while operating under the following conditions:

(a) Rated speed.
(b) Rated pressure.
(c) Inlet oil temperature 82°C (180°F).

The measured overall sound power level of the pump should not exceed W dBA re 10^{-12} watt, where $W = 10 \log_{10}(hp) + 74$

$$hp = \frac{\text{Rated pressure (psi) x Flow at rated speed (gpm)}}{1714}$$

4.4.3 Pump number two should be subjected to the test described in section 4.4.3.1.

4.4.3.1 Contaminant sensitivity: Flow degradation of the pump when exposed to particulate contaminants should be measured and reported in accordance with NFPA T3.9.18 under the following test conditions:

(a) Rated speed
(b) Rated pressure
(c) Inlet oil temperature 82°C (180°F)

Flow degradation should be 0% after the 0-10 μm injection and not more than 30% after the 0-40 μm injection.

4.4.4 Pumps number three, four, and above, if applicable, should be subjected to the test described in section 4.4.4.1.

4.4.4.1 Durability: The pumps should be subjected to the durability test specified in OSU-P-4-May 1979. The minimum number of cycles each pump must erdure is dependent upon the number of pumps to be tested, as set forth in the following table:

N - Number of pumps tested	Minimum number of cycles for each pump
2	1,500,000
3	900,000
4	650,000
5	500,000

Volumetric efficiency should be measured at intermediate values of 500,000, 650,000, 900,000, and 1,500,000 cycles in accordance with the test procedure. Minimum acceptable volumetric efficiency is 75%.

5. PACKAGING

5.1 Preservation: Preservations are level A or commercial as specified in 6.

5.1.1 Level A: The interior surfaces of the pump are coated with preservative lubricating oil MIL-H-6083. Each pump is then wrapped with greaseproof barrier-material conforming to MIL-B-121, type I, grade A, class 2. The wrap is secured with tape conforming to PPP-T-60, type III, class 1. Each pump is placed in a box conforming to PPP-B-636, W5c. Cushioning is provided to prevent movement within the box. The box is closed and sealed as specified for method V in the appendix to the box specification.

5.1.2 Commercial: Each pump is preserved in accordance with MIL-STD-1188.

5.2 Packing: The packing is level A or commercial as specified in 6.

5.2.1 Level A: The pump, preserved as required in 5.1, is packed in close-fitting boxes conforming to PPP-B-601, overseas type, optional style. The boxes are closed strapped in accordance with the appendix to the box specification. Strapping conforms to QQ-S-781, class I, type I or IV, size as applicable and unless otherwise specified in 6, is finish B.

5.2.2 Commercial: The pump preserved as required in 5.1 is packed in accordance with MIL-STD-1188.

5.3 Marking

5.3.1 Military: Marking for military levels of protection is in accordance with MIL-STD-129.

5.3.2 Commercial: Marking for commercial packaging is in accordance with MIL-STD-1188.

6. ORDERING DATA: Procurement documents will specify the following:

(a) Title, number and date of this specification.
(b) Type of pump required.
(c) Rated pressure required.
(d) Rated spped required.
(e) Minimum operating speed required.
(f) Volumetric displacement required.
(g) Direction of shaft rotation.
(h) Level of preservation required.
(i) Level ofpacking required.
(j) Number of pumps to be packed together.
(k) Government part number when specified by end item drawing.

7. QUALITY ASSURANCE PROVISIONS

7.1 Inspection responsibility: Unless otherwise specified in the contract or purchase order, the pump manufacturer is responsible for all quality assurance inspections and tests. Except as may otherwise be specified in the contract order, the pump manufacturer may use his own or other suitable facilities for conducting performance testing and inspections required herein. The government reserves the right to reject these facilities as unsuitable to assure supplies and services are as required herein.

7.2 Inspection classification: Inspections are classified as:

(a) Quality confromance inspection (7.3)
(b) Quality performance inspection (7.4)
(c) Inspection of packaging (7.5)

7.3 Quality conformance inspection: The pump will be examined for conformance with the requirements of this specification. The presence of one of the following defects is cause for rejection:

101. Type not as identified on data plate.
102. Markings illegible or missing.
103. Material not new or as specified.
104. Dissimilar metal not used as required.
105. Direction of rotation not as identified.
106. Mounting flange not as required.

107. Ports not as required.
108. Seals not as specified.
109. Indentification Plate not as required.
110. Workmanship not as required.
111. Technical Data missing or not as required.
112. QA inspection and testing not as required.

7.4 Quality performance inspection

7.4.1 Test Data: The manufacturer will test the pump as specified herein and present performance data for the following tests.

(a) Low temperature test (4.4.2.1)
(b) Performance and efficiency (4.4.2.2)
(c) Pump noise generation (4.4.2.3)
(d) Contaminant sensitivity (4.4.3.1)
(e) Durability test (4.4.4.1)

7.4.2 Production tests: Each pump should be tested at rated speed and 1.2 times rated pressure for not less than 5 minutes. The pump should show no evidence of leakage and volumetric efficiency at rated speed and rated pressure should be at least 75%.

7.5 Inspection of packaging

7.5.1 Units of pack: For the purpose of inspection, a completed pack, prepared for shipment, is considered a unit of pack.

7.5.2 Sampling: Sampling for examination is conducted in accordance with MIL-STD-105.

7.5.3 Examination: Selected samples are examined for the following defects with AWL 2.5 percent defective.

113. Materials and containers are not as specified. Each incorrect material or container is one defect.
114. Interior surfaces of the pump are not coated with preservatives as specified on level A or commercial.
115. The pump is not wrapped with greaseproof paper as required for level A.
116. Greaseproof wrap is not secured with tape as required for level A.
117. Cushioning is not provided to prevent movement of the pump within the container as required for level A.
118. Strapping is not as required for level A.
119. Preservation and packing are not in accordance with referenced documents as required for commercial.
120. Markings are illegible, incomplete, or incorrect as required for level A or commercial.
121. Number of pumps to be packed together are not as required for level A or commercial.

8. QUALIFICATION

8.1 Unless otherwise specified, when the pump covered by this specification is procured as an end item by the Government, only bids or proposals offering a pump which has been certified by the manufacturer under this specification at the time set for opening of the bids or award of a negotiated contract will be considered in making award.

8.2 When the pump covered by this specification is a component of an end item being procured by the Government, the pump supplier will be notified of the requirements of this specification in accordance with ASPR 1-1107.2(b).

8.3 The attention of Original Equipment Manufacturers (OEM) is called to the above conditions and their suppliers are urged to self certify pumps they propose to offer to the Government in order that they may be eligible to be awarded contracts or orders for pumps covered by this specification.

9. DATA REQUIREMENTS: Unless otherwise specified, the contracting officer will require pump manufacturers certification data specified herein be furnished to the Government with each bid or proposal offering. Additionally, the contracting officer may require such data as technical publications, instructional materials, illustrated parts list, maintenance and operation manuals be furnished with each pump.

Custodian:
Army-ME

Preparing activity:
Army-ME
Project No. 4320-

User interest:
Army-AT

APPENDIX C

METHOD FOR ESTABLISHING THE DURABILITY OF A FIXED DISPLACEMENT FLUID POWER PUMP

(OSU-P-4-May 1979)

1. SCOPE To include a method for determining the durability of a fixed displacement, fluid power pump.

2. PURPOSE To verify the ability of a pump to perform satisfactorily during a specified period of time when subjected to cyclic discharge pressure at specified conditions of temperature, shaft speed, and system fluid.

3. TERMS

3.1 Pump Durability The ability of a pump to endure specified operating conditions for an extended period of time.

3.2 For definitions of other fluid power terms used herein, see ANSI/B93.2.

4. GENERAL PROCEDURE

4.1 Install pump in a test system as shown in Fig. C-1.

4.2 Measure initial volumetric efficiency per Section 6 and record in the Test Data Sheet.

4.3 Operate pump per Section 7 for the number of cycles specified. At specified intermediate values of life, halt durability test and measure volumetric efficiency per Section 6. Record volumetric efficiency in the Test Data Sheet.

4.4 Measure final volumetric efficiency per Section 6.

4.5 Reject the pump if volumetric efficiency becomes less than the specified minimum volumetric efficiency prior to the completion of the specified minimum number of cycles or if leakage from the pump shaft exceeds 1 drop per 10 minutes.

4.6 Repeat steps 4.1 to 4.5 for each of the remaining pumps.

4.7 Present data per Section 8.

5. TEST EQUIPMENT

5.1 Circuit: Use a test circuit as shown in Fig. C-1.

5.2 Oil: Use oil specified by durability test sponsor.

5.3 Filtration: The control filter will limit the total number of particles in the system fluid to 1000 particles per millilitre greater than 10 microns. Greater cleanliness may be necessary for some pumps to insure against contaminant wear failure rather than fatigue failure.

5.4 Aeration: Minimize fluid aeration by taking precautions such as proper system design and by adequate removal of air from the system before testing. The inlet oil must be visually free of entrained air throughout testing.

5.5 Tolerance of Test Parameters: Maintain the test parameters within the limits set forth in Table C-1 unless otherwise specified in the following test procedures.

TABLE C-1. Test Parameter Accuracy.

Test Condition	SI Unit	Inch Unit	Maintain Within (±) of Actual
Shaft Speed	rpm	rpm	2%
Pressure below atmospheric	bar	in Hg	1 in Hg
Atmoshperic pressure & greater	bar	psi	2%
Fluid Temperature	^{o}C	^{o}F	$2.8^{o}C(5^{o}F)$

5.6 Inlet Oil Temperature: Maintain inlet oil temperature at $180^{o}F$ unless specified otherwise by durability test sponsor.

5.7 Inlet Pressure: Maintain inlet oil pressure at atmospheric pressure ±1 in Hg unless otherwise specified by durability test sponsor.

5.8 Run-In: Before the tests are carried out, "run-in" the pump in accordance with manufacturer's recommendations.

5.9 Instrumentation Accuracy: Select and maintain instrumentation so that it is accurate within the limits set forth in Table C-2, unless otherwise specified in the following test procedures.

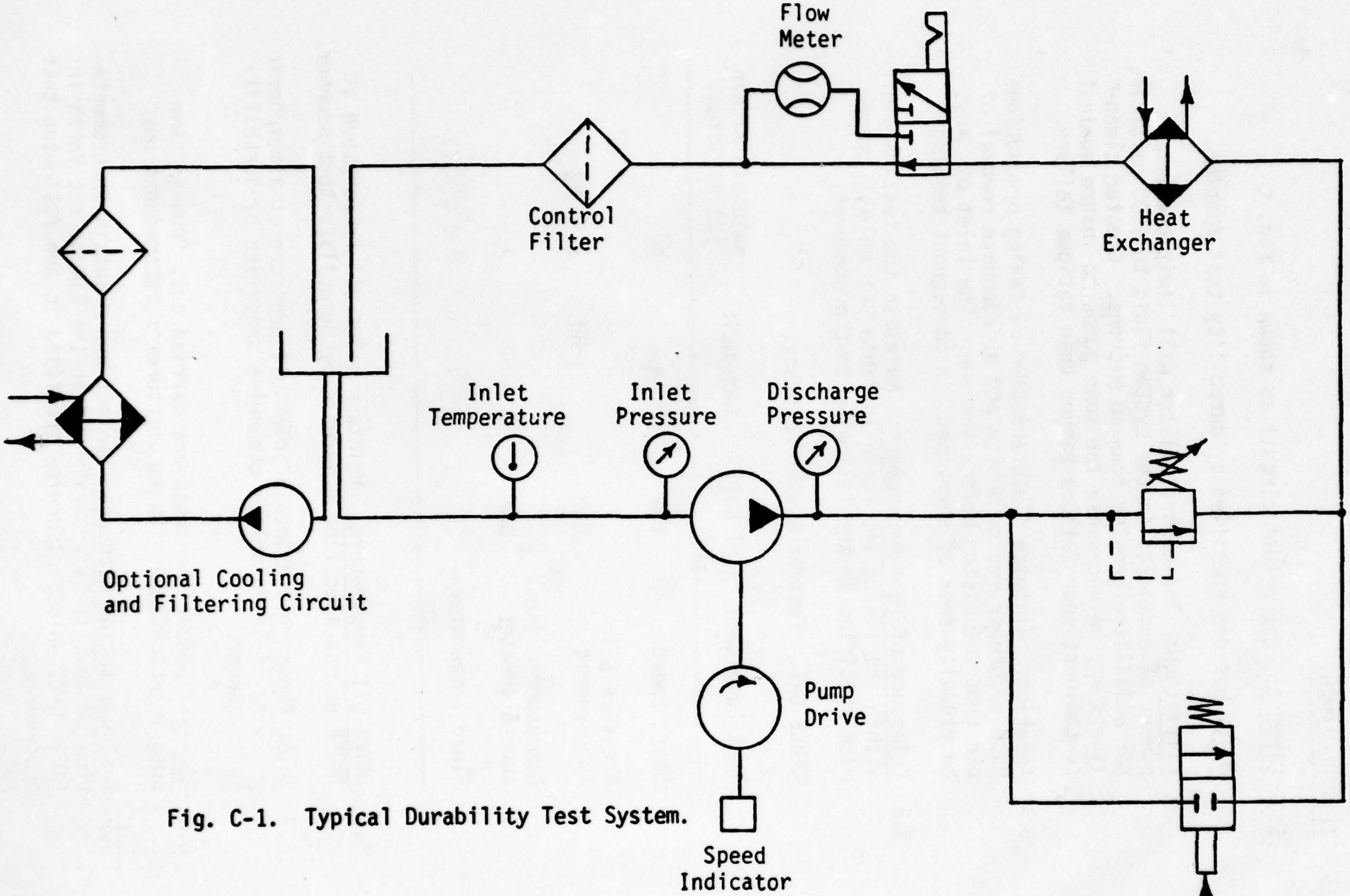

Fig. C-1. Typical Durability Test System.

TABLE C-2. Instrumentation Accuracy.

Quantity	SI Unit	Inch Unit	Accuracy Within (±) of Actual
Pressure below atmoshperic	bar	in Hg	0.5 in Hg
Atmospheric pressure & greater	bar	psi	2%
Flow	ℓ/min	USGPM	2%
Shaft speed	rpm	rpm	0.5%
Temperature	^oC	^oF	$0.5^oC(1^oF)$

6. <u>VOLUMETRIC EFFICIENCY TEST</u>

6.1 Operate the pump at manufacturer's rated pressure and rated speed.

6.2 Measure and record output flow rate.

6.3 Using the measured value of flow rate and the pump manufacturer's values for rated speed and volumetric displacement, calculate volumetric efficiency:

$$\text{Volumetric Efficiency} = \frac{\text{Q Measured}}{\text{Volumetric Displacement x Rated Speed}}$$

7. <u>DURABILITY TEST</u>

7.1 Operate the pump under the following conditions:

7.1.1 <u>Speed</u>: Manufacturer's Rated Speed +0%/-20%.

7.1.2 <u>Outlet Pressure</u>: Cycled between 5% of manufacturer's rated pressure (±5% of RP) and 120% of manufacturer's rated pressure (±5% of RP).

7.1.3 Endurance Cycle Conditions: 60 ± 3 cycles/minute (except as otherwise noted; waveform is to be consistent with the impulse test waveform of SAE J343).

8. <u>DATA PRESENTATION</u>

8.1 Provide all information in the pump durability test data sheet shown in Fig. C-2.

Fig. C-2. Pump Durability Test Data Sheet.

PUMP

MFR. NAME____________________

MFR. PART NO.____________________

MFR. SERIAL NO.____________________

MFR. RATED PRESSURE____________________

MFR. RATED SPEED____________________

DIRECTION OF ROTATION____________________

MFR. VOLUMETRIC DISPLACEMENT____________________

DURABILITY TEST

DATE OF TEST____________________

TEST LOCATION____________________

FLUID____________________

INLET OIL TEMPERATURE____________________

INLET OIL PRESSURE____________________

NUMBER OF PUMPS TESTED____________________

		VOLUMETRIC EFFICIENCY				
PUMP NO.	BEGINNING TEST	* CYCLES	CYCLES	CYCLES	CYCLES	END OF TEST
1						
2						
3						
4						
5						

FINAL RATE OF LEAKAGE   Pump No. 1________   Pump No. 2________

Pump No. 3________   Pump No. 4________   Pump No. 5________

*Enter above columns the intermediate values of life at which volumetric efficiency is to be measured.

APPENDIX D

METHOD FOR ESTABLISHING THE LOW TEMPERATURE OPERATION ABILITY OF A FIXED DISPLACEMENT, FLUID POWER PUMP

(OSU-P-6 May 1979)

1. SCOPE To include a method for determining the ability of a fixed displacement fluid power pump to survive under low temperature conditions.

2. PURPOSE To verify the ability of a pump to withstand low temperature operation and perform satisfactorily at specified conditions of speed, discharge pressure and system fluid.

3. TERMS

3.1 For definitions of fluid power terms used herein, see ANSI/B93.2.

4. TEST EQUIPMENT

4.1 Test environment: Use a cold room capable of maintaining air temperature of -54 ± 5°C (-65 ± 5°F).

4.2 Circuit: Use a fluid power test circuit as shown in Fig. D-1. The pump drive and the clean-up filter may be located external to the test environment.

4.3 Oil: Use low temperature oil as specified.

4.4 System volume: Adjust the total system volume to be equal to the volume of fluid delivered by the pump when operated at rated speed for 30 seconds.

4.5 Filtration: The "clean-up" filter will limit the total number of particles in the system fluid to 1000 particles per millilitre greater than 10 microns.

4.6 Aeration: Minimize fluid aeration by taking precautions such as proper system design and by adequate removal of air from the system before testing. The inlet oil should be visually free of entrained air throughout testing.

4.7 Tolerance of test parameters: Maintain the test parameters within the limits set forth in Table D-1 unless otherwise specified in the following test procedures.

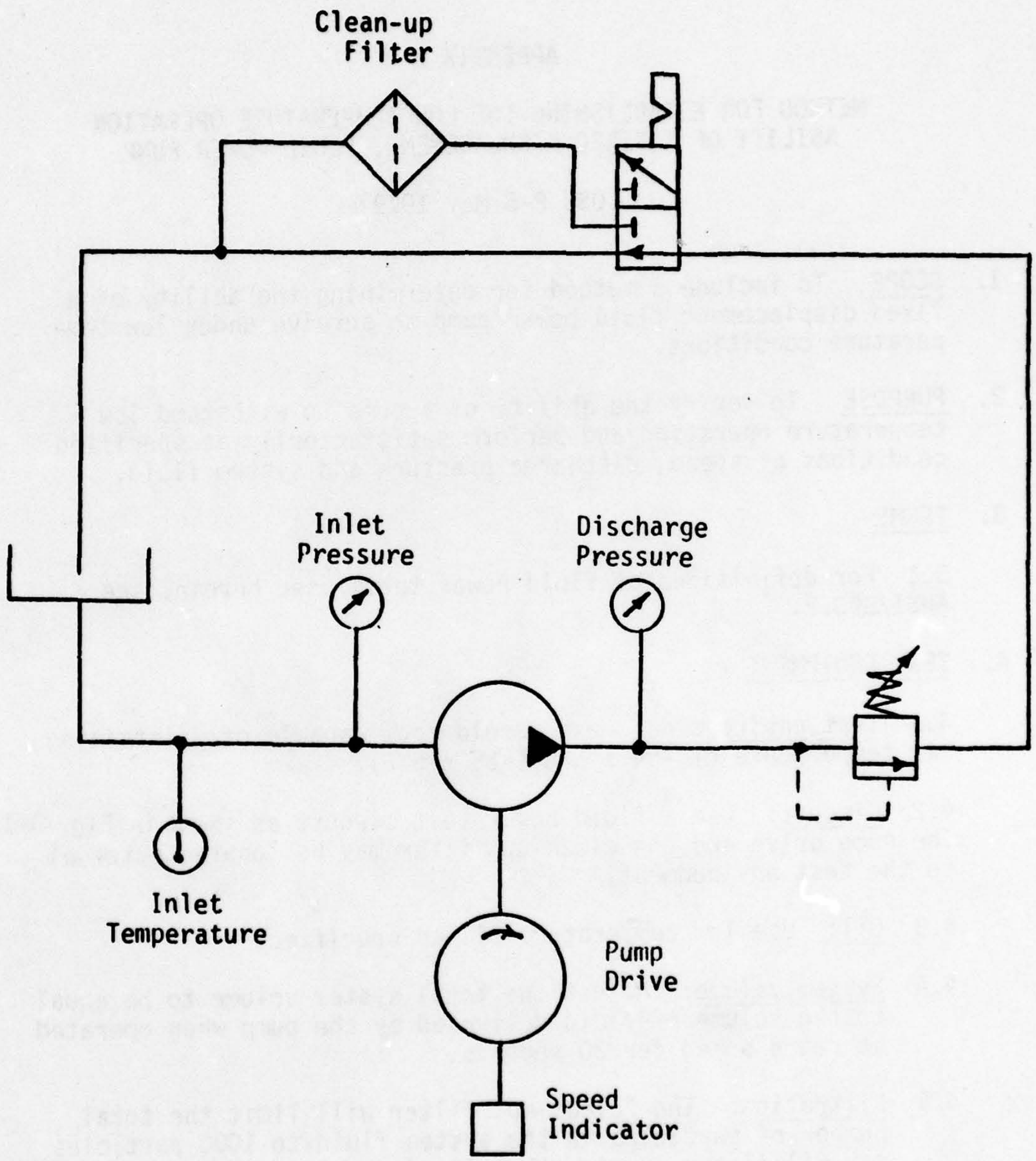

Fig. D-1. Typical Low Temperature Test System.

TABLE D-1. Test Parameter Accuracy.

Test Condition	SI Unit	Inch Unit	Maintain Within (±) of Actual
Shaft Speed	rpm	rpm	2%
Pressure Below Atmospheric	bar	in Hg	1 in Hg
Atmoshperic Pressure and Greater	bar	psi	2%
Fluid Temperature	^{o}C	^{o}F	$2.8^{o}C$ ($5^{o}F$)

4.8 Inlet Pressure: Maintain inlet oil pressure at atmospheric pressure ±1 in Hg unless otherwise specified by durability test sponsor.

4.9 Run-In: Before the tests are carried out, "run-in" the pump in accordance with manufacturer's recommendations.

4.10 Instrumentation Accuracy: Select and maintain instrumentation so that it is accurate within the limits set forth in Table D-2 unless otherwise specified in the following test procedures.

TABLE D-2. Instrumentation Accuracy.

Quantity	SI Unit	Inch Unit	Accuracy Within (±) of Actual
Pressure Below Atmospheric	bar	in Hg	0.5 in Hg
Atmoshperic Pressure and Greater	bar	psi	2%
Flow	ℓ/min	USGPM	2%
Shaft Speed	rpm	rpm	0.5%
Temperature	^{o}C	^{o}F	$0.5^{o}C$ ($1^{o}F$)

5. PROCEDURE

5.1 Install the pump in the test circuit.

5.2 Circulate system fluid through "clean-up" filter for 15 minutes.

5.3 Bypass "clean-up" filter.

5.4 Lower the temperature of the cold room to -54 ± 3°C (-65 ± 5°F) and maintain this temperature for a period of at least 12 hours.

5.5 Raise the temperature of the cold room to -46 ± 3°C (-50 ± 5°F).

5.6 When the temperature of the fluid in the reservoir reaches -46°C (-50°F), perform the following operations (See Fig. D-2.):

5.6.1 Start-up the pump by gradually (< 15 sec.) raising the speed to 30% of rated speed and outlet pressure to 10% of rated pressure.

5.6.2 When the inlet temperature reaches -26°C(-15°F), gradually (< 12 sec.) raise pump speed to rated speed.

5.6.3 When the inlet temperature reaches -21°C(-5°F), gradually (< 6 sec.) raise outlet pressure to rated pressure.

5.6.4 When the inlet temperature reaches -15°C (+5°F), terminate test.

5.7 Record any malfunction or shaft leakage in excess of 1 drop per 10 minutes.

6. <u>DATA PRESENTATION</u>

6.1 Provide all information in the low temperature test data sheet shown in Fig. D-3.

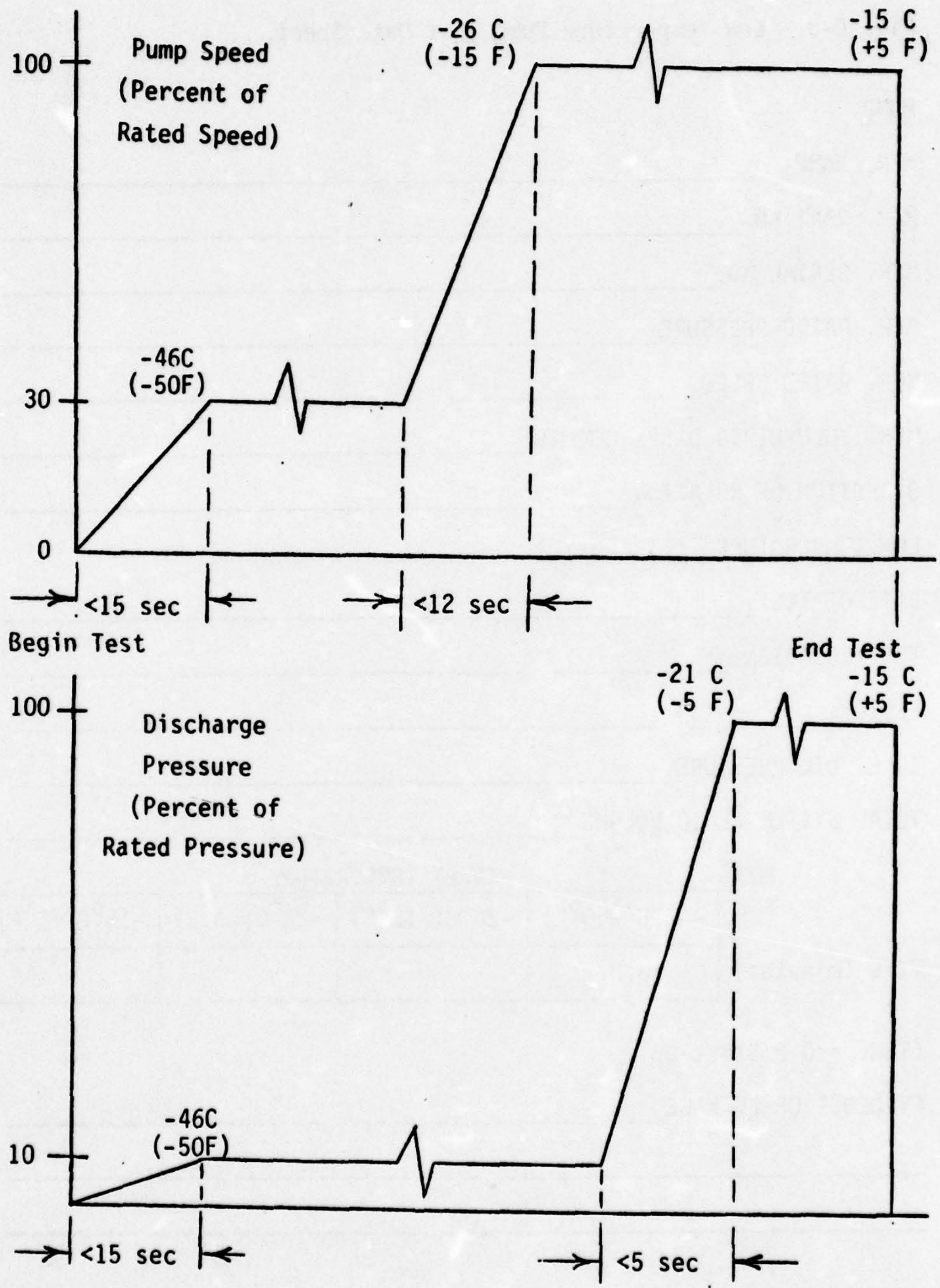

Fig. D-2. Low Temperature Test Schedule.

Fig. D-3. Low Temperature Pump Test Data Sheet.

PUMP

MFR. NAME ____________________

MFR. PART NO. ____________________

MFR. SERIAL NO. ____________________

MFR. RATED PRESSURE ____________________

MFR. RATED SPEED ____________________

MFR. VOLUMETRIC DISPLACEMENT ____________________

DIRECTION OF ROTATION ____________________

LOW TEMPERATURE TEST

DATE OF TEST ____________________

TEST LOCATION ____________________

FLUID ____________________

INLET OIL PRESSURE ____________________

TOTAL SYSTEM FLUID VOLUME ____________________

	INLET TEMPERATURE			
	$-46^{o}C(-50^{o}F)$	$-26^{o}C(-15^{o}F)$	$-21^{o}C(-5^{o}F)$	$-15^{o}C(+5^{o}F)$
TIME (minutes)	0.0			

(TIME = 0 @ Start-Up)

EVIDENCE OF LEAKAGE ____________________

____

____

APPENDIX E

PRODUCTION QUALITY CONTROL FATIGUE TESTING WITH SMALL SAMPLE SIZES

One "dream" of some manufacturers might likely be to obtain 100% confidence in predictions based on the results of a test on a single sample. Most organizations recognize the "dream" for what it is and adjust their test programs accordingly. These companies invest their dollars in assuring quality which allows them to minimize their expenditure for liability insurance while keeping customers pleased with the reliability of their products. The current rash of liability suits shows that the consumer desires the latter, or at least respects the latter, because in the latter case there are fewer opportunities to file lawsuits.

Assuring a given level of consumer risk requires a commitment to invest in testing. Once that commitment is made, it is desirable to get the most reliability information for the least number of dollars. This paper discusses a quality control technique which allows the consumer to establish his level of risk and the producer to sequentially test random samples from a Production Lot until it is obvious the lot is acceptable or until the producer decides to stop the tests and reject the lot. An example illustrates the procedure for a Production Lot sample size of five and a consumer risk of 10%. The procedure uses the Weibull domain as the reference for establishing and verifying acceptable lots. The example shows how to develop a Decision Table based on the results of Design Acceptance Testing

on five units. The Decision Table is used by the producer to accept or reject the subsequent production lots.

ACCEPTANCE TESTING

Quality control is defined as [1]:

> "Quality control is the regulatory process through which we measure actual quality performance, compare it with standards, and act on the difference."

In general, it is possible to divide the quality control function into two distinct areas—Design Acceptance Testing and Production Lot Acceptance Testing. Design Acceptance Testing might be thought of as "Validation Testing" or "Quality Establishment." Design Acceptance Testing is the process of verifying through testing that a given design meets (or exceeds) a user's requirements. Production Lot Acceptance Testing might be though of as "In-Process Testing." Production Lot Acceptance Testing is the process of verifying that a given production lot meets, with a specified consumer risk, the user's requirements. A "Production Lot" is purportedly from a "qualified" design whose cumulative failure distribution is known. Figure E-1 illustrates the relationship between Quality Control, Design Acceptance, and Production Lot Acceptance.

ACCEPTANCE SAMPLING RISKS

In the process of either Design Acceptance or Production Lot Acceptance, we want to make the correct assessment; that is, only select "good" products. Associated with any decision is a finite amount of risk, a chance that we will make the wrong decision. That is, we may accept a "bad" product. The chance of getting a

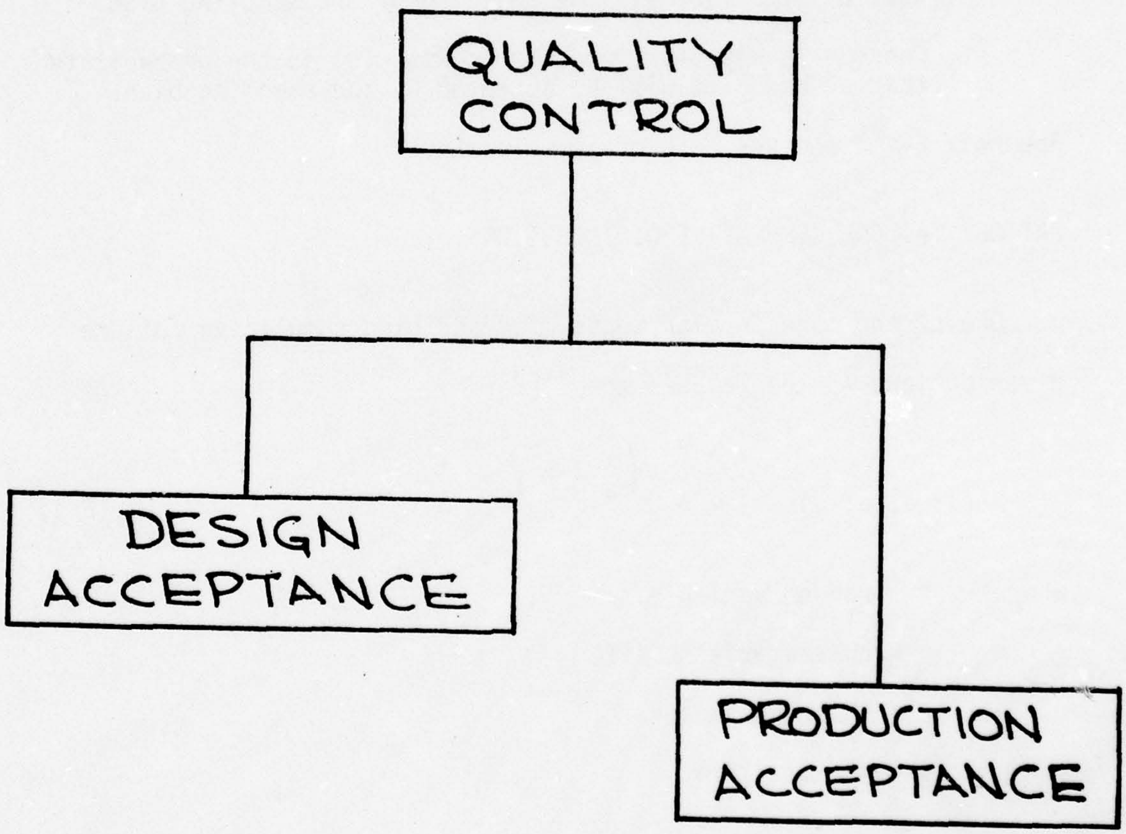

Fig. E-1. Schematic of Two Areas of Quality Control—Design Acceptance and Production Lot Acceptance.

bad product is referred to as user or consumer risk. Although we would like for the user's risk to be zero, for reasons of practicality, we must settle on some level of risk. The risks associated with acceptance sampling are defined as [1]:

> Producer's Risk—The Producer's Risk (α) is the probability that a "good" lot will be rejected by the sampling plan.
>
> Consumer's Risk—The Consumer's Risk (β) is the probability that a "bad" lot will be accepted by the sampling plan.

Appendix E-A discusses both of these risks.

FATIGUE FAILURE CUMULATIVE DISTRIBUTIONS

One of the more popular means of describing cumulative failure distributions is the Weibull model [2]:

$$F\ (X;\ \theta,\ \beta,\ \delta) = 1 - e^{-\left(\frac{X - \delta}{\theta - \delta}\right)^{\beta}} \qquad X \geq \delta \qquad \text{(E-1)}$$

where: X = random variable

θ = characteristic life

β = shape parameter

δ = minimum life

Equation (1) can be rearranged to yield:

$$\ell n\ (\ell n\ (1/(1 - F(x))) = \beta\ \ell n\ (X - \delta) - \beta\ \ell n\ (\theta - \delta) \qquad \text{(E-2)}$$

Figure E-2 shows the Weibull domain described by Eq. (E-2), where $\ell n\ (\ell n\ (1/1 - F(x)))$ is plotted versus $\ell n\ x$.

Appendix E-B, Weibull Failure Analysis Chart, summarizes the relationships needed to plot failure data on Weibull Probability

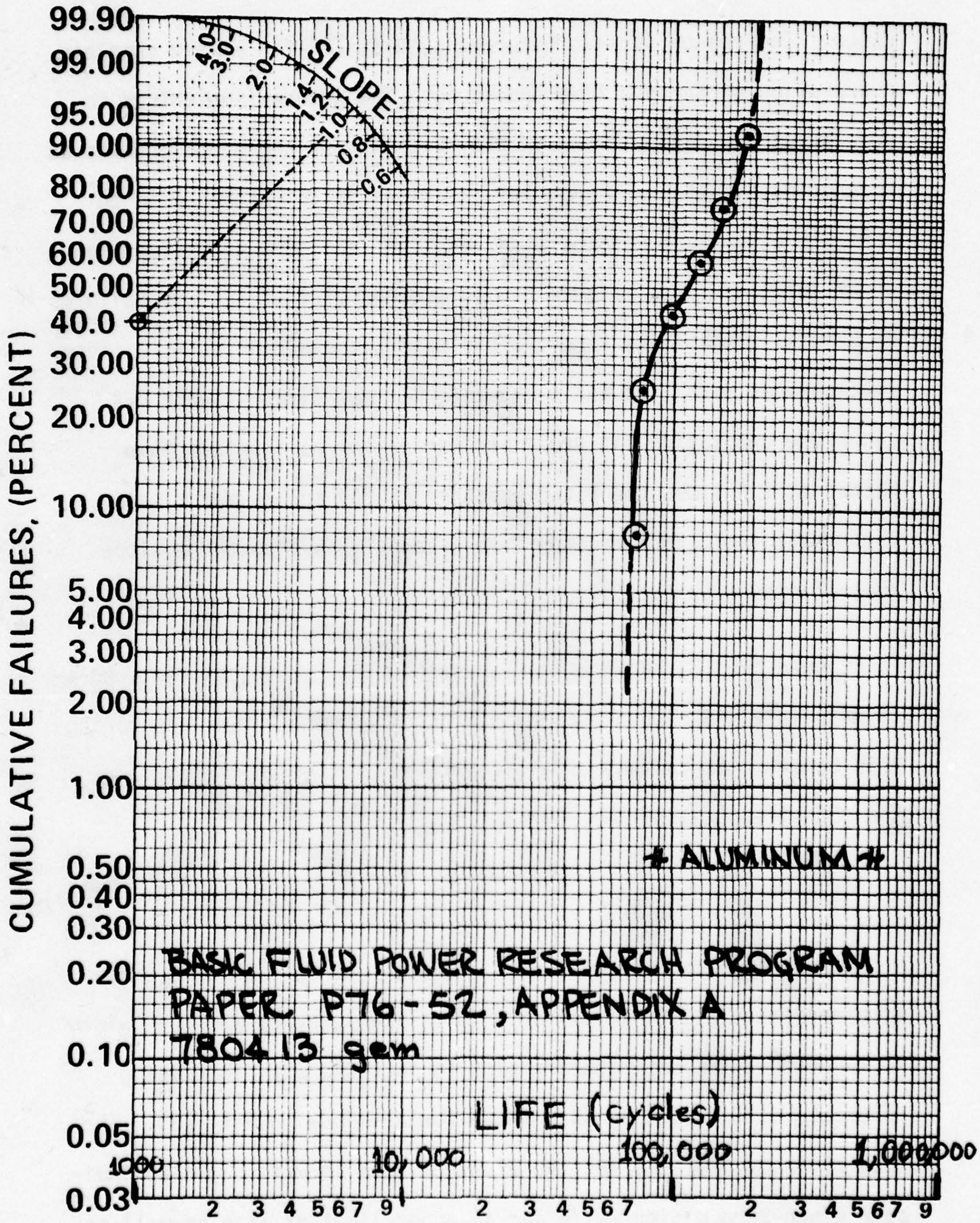

Fig. E-2. Cumulative Failure Distribution for Aluminum Showing Changing Slope and Non-Zero Minimum Life.

Paper. The reader is also referred to Ref. [2] for a more complete description of the plotting procedure. The interesting things to note about the data plotted in Fig. E-2 is that the slope changes as a function of life, the cumulative failure distribution seems to indicate a non-zero minimum life, and the data seem to indicate an upper limit of life. Figure E-3 is another plot of fatigue failures on Weibull Probability Paper. This data set also indicates a slope that changes with life and a non-zero minimum life. Figure E-4 shows the cumulative failure distribution for an aluminum pump tested at 1.2 times the rated pressure at 360 cycles per minute. These data indicate a minimum life greater than zero and a slope that changes with life.

Figure E-5 [3] shows that, for steel, the "width" of the cumulative failure distribution changes with life. One way of describing this behavior is the use of a life ratio, L*. L* is the ratio of the 90% Life, L_{90}, to the 10% Life, L_{10}.

$$L^* = L_{90}/L_{10} \qquad \text{(E-3)}$$

Note that, for a stress of 1.0, L* is about 52, while for a stress of approximately 1.2, L* is 3.8. This same tendency for the "width" of the cumulative distribution plot to increase with increasing "Characteristic Life" is also shown for aluminum in Ref. [3]. The data plots of Fig. E-5 and the aluminum data in Ref. [3] also indicate a non-zero minimum life and an upper limit of life as well as the changing slope as a function of life.

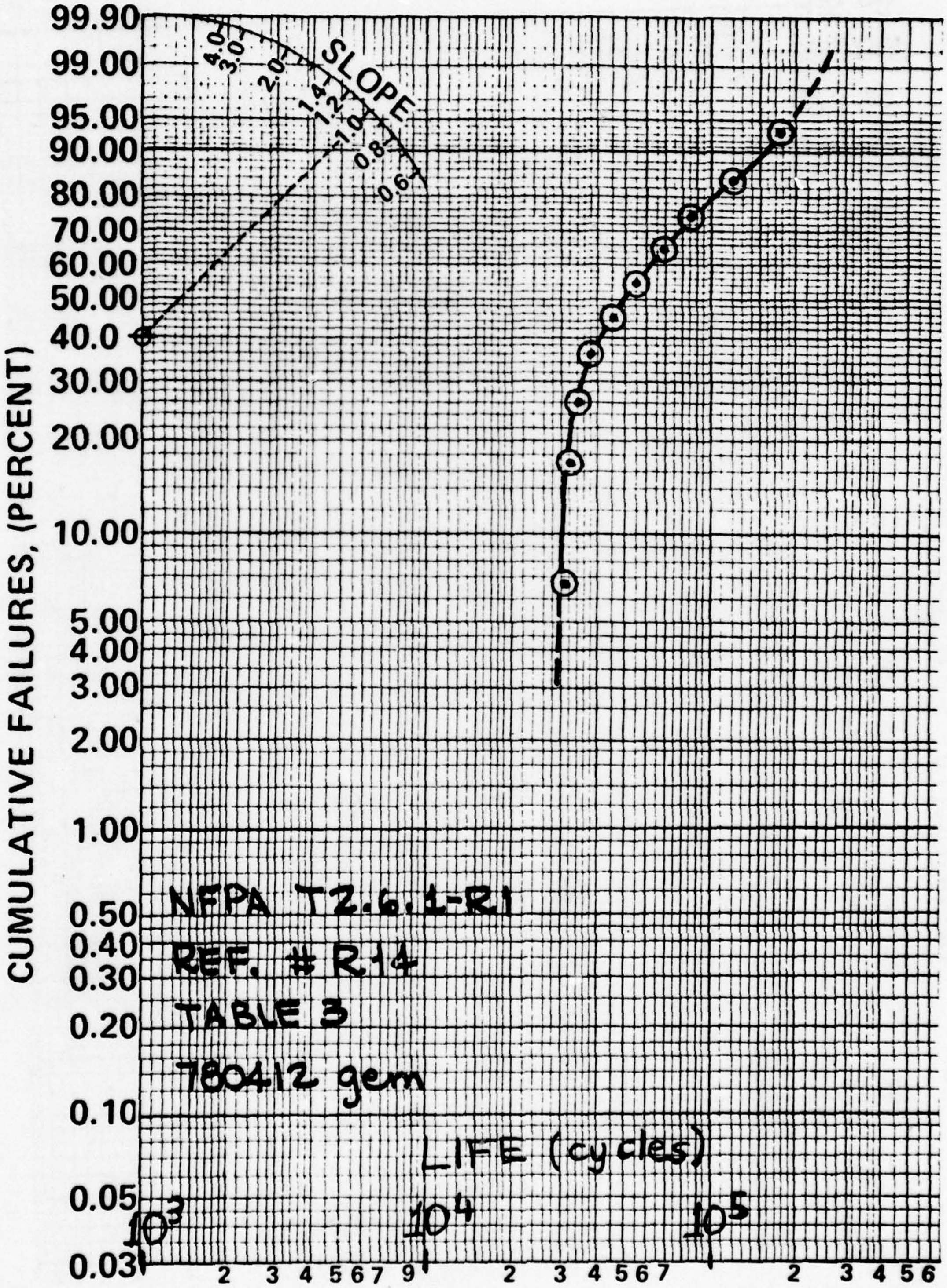

Fig. E-3. Failure Distribution Showing Change in Slope as a Function of Life.

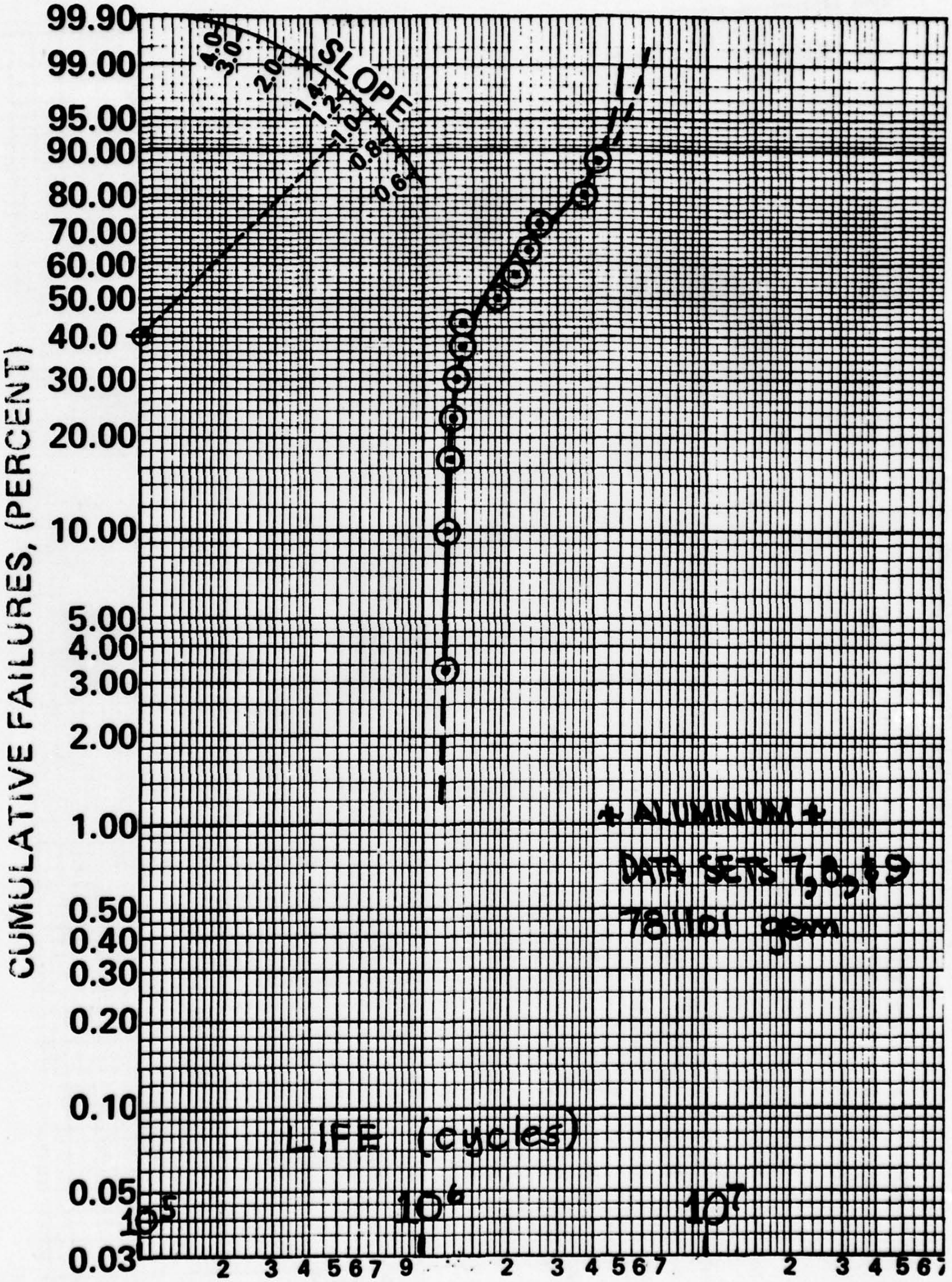

Fig. E-4. Cumulative Failure Distribution Showing Non-Zero Minimum Life and Finite Life.

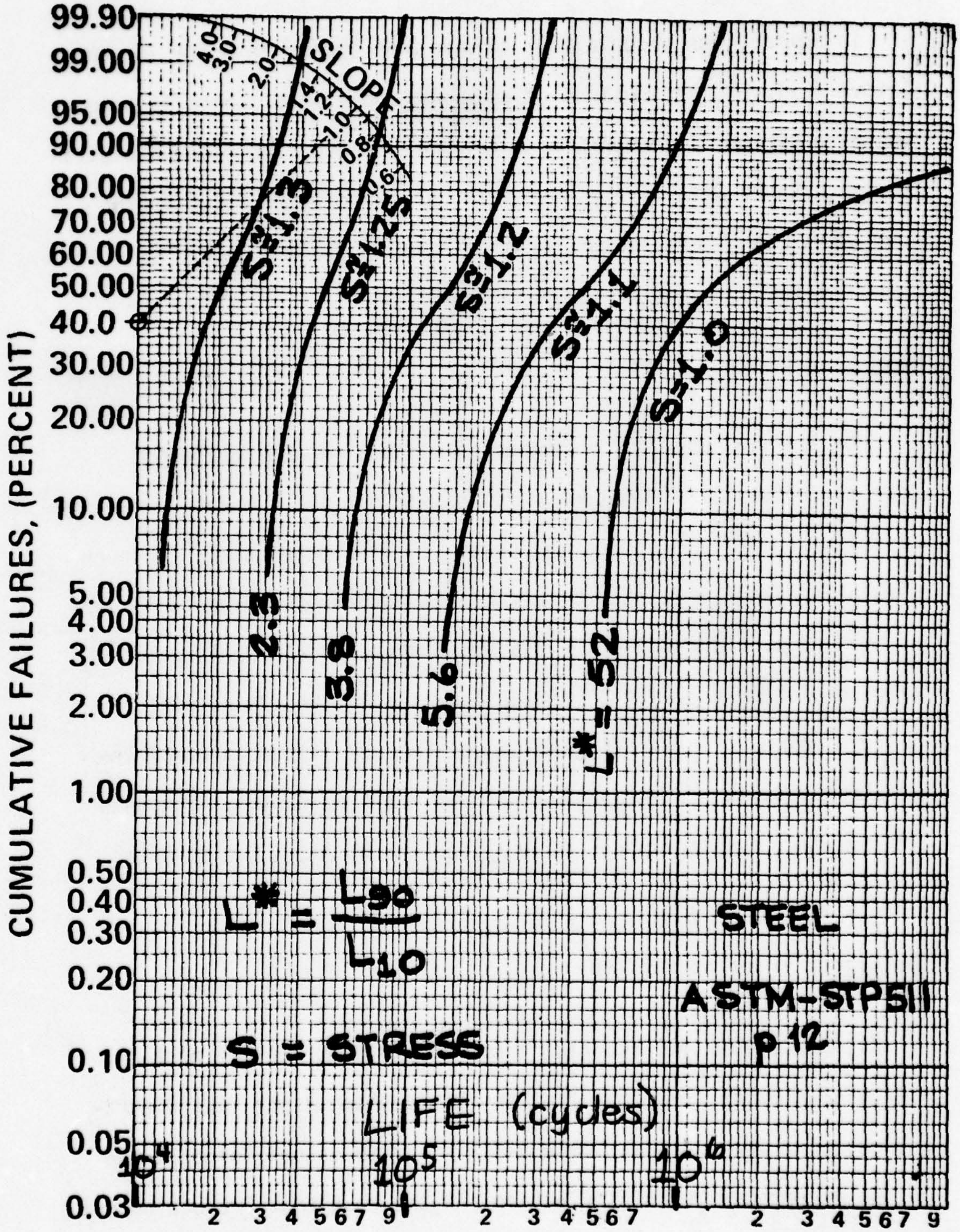

Fig. E-5. Failure Distribution Showing Increasing "Curve Width" with Increasing Characteristic Life.

In summary, we observe four things that appear to occur in cumulative failure distributions of aluminum and steel: There is some non-zero minimum life for all units; the slope of the failure distribution changes with life; the "width" of the failure distribution increases with increasing "characteristic life"; and, there is some finite life for the distribution.

DESIGN ACCEPTANCE TESTS

For the purposes of this paper, it is assumed that a Design Acceptance Test has been conducted and yielded the cumulative failure distribution for the design in question. This means that, for each cumulative failure point, the associated number of cycles to failure is available. Figure E-6 shows a minimum acceptable performance point depicted as a life of 500,000 cycles at the 10% cumulative failure point. Figure E-6 also shows the curve fitted plot of the cumulative failure distribution for the verified design. The data for the Design Acceptance Test are given in Appendix E-C as Data Set 7.

PRODUCTION ACCEPTANCE TESTS

The production lot acceptance cumulative distribution plot is established using the "shape" of the Design Acceptance Test Distribution. For the example, Table E-1 lists life values for both the Design Acceptance Test Distribution and the resultant Production Lot Acceptance Test Distribution. The Production Lot lives are obtained by using the desired life of 500,000 cycles at 10% cumulative failures as a reference point and using the "shape" of the Design

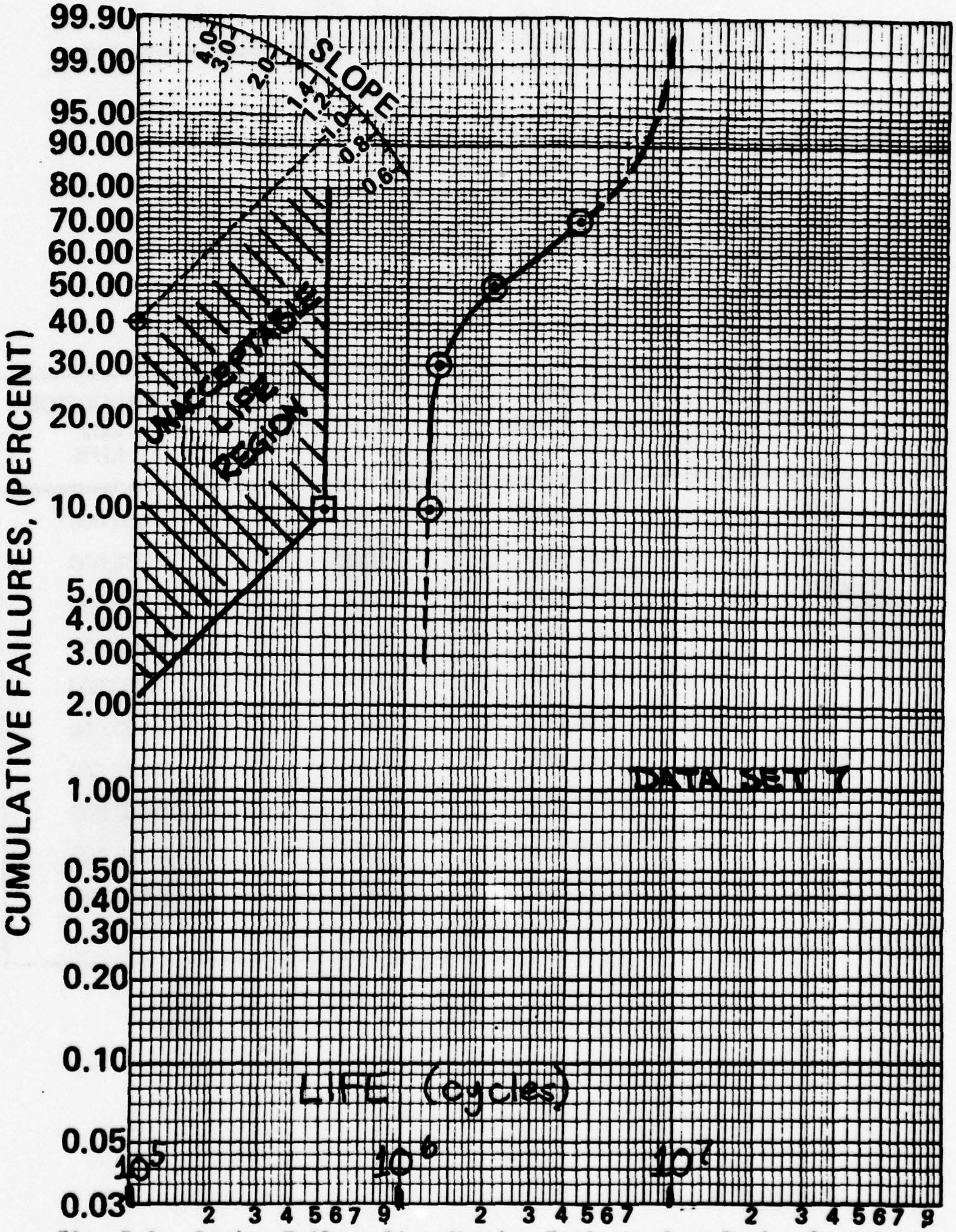

Fig. E-6. Design Failure Distribution Estimate from Design Acceptance Testing.

TABLE E-1. Production Lot Acceptable Life Calculations Based on Design Acceptance Lives.

CUMULATIVE % FAILURES	DESIGN ACCEPTANCE LIFE	LIFE RATIO L_i / L_{10}	PRODUCTION LOT LIFE
10.0	1,200,000	1.00	500,000
36.9	1,500,000	1.25	625,000
43.8	1,700,000	1.42	710,000
53.6	2,400,000	2.00	1,000,000
68.4	4,200,000	3.50	1,750,000
90.0	7,500,000	6.25	3,125,000
94.9	8,100,000	6.75	3,375,000
96.5	8,500,000	7.08	3,540,000
97.4	8,700,000	7.25	3,625,000
97.9	8,800,000	7.33	3,665,000

Acceptance sample to obtain all other Production Lot acceptable lives. Figure E-7 shows the plot of the "acceptable" production lot failure distribution.

We are now in the position to establish Production Lot acceptance criteria. Note that, for the distribution shown in Fig. E-7, there is only a 10% chance of the first pump tested lasting longer than 3,125,000. This figure is obtained by reading the graph at the 90% cumulative failure point. So, if the first pump tested lasts longer than 3,125,000 cycles, there is less than a 10% chance that it is from a distribution that is "worse" than the one depicted in Fig. E-7. In other words, there is less than a 10% chance we would accept a bad lot by accepting the lot if the first pump lasts longer than 3,125,000 cycles.

Now, if the first pump fails before 3,125,000, how do we establish "pass" criteria? First, we ask the Question: "If two pumps are tested, what probability, F(x), is associated with a 10% chance that both pumps are from the acceptable distribution or a better distribution?" To answer our question, we must find the probability, F(x), such that doubling the number of samples we still only have a consumer's risk of 10%. What we want then is some F(x) such that [4]:

$$F(x)^2 = 0.9 \qquad \text{(E-4)}$$

or $$F(x) = 0.9^{\frac{1}{2}} = 94.9\% \qquad \text{(E-5)}$$

For two samples, the cumulative percent failure point is 94.9%. In other words, we have a 94.4% chance of the first unit failing before

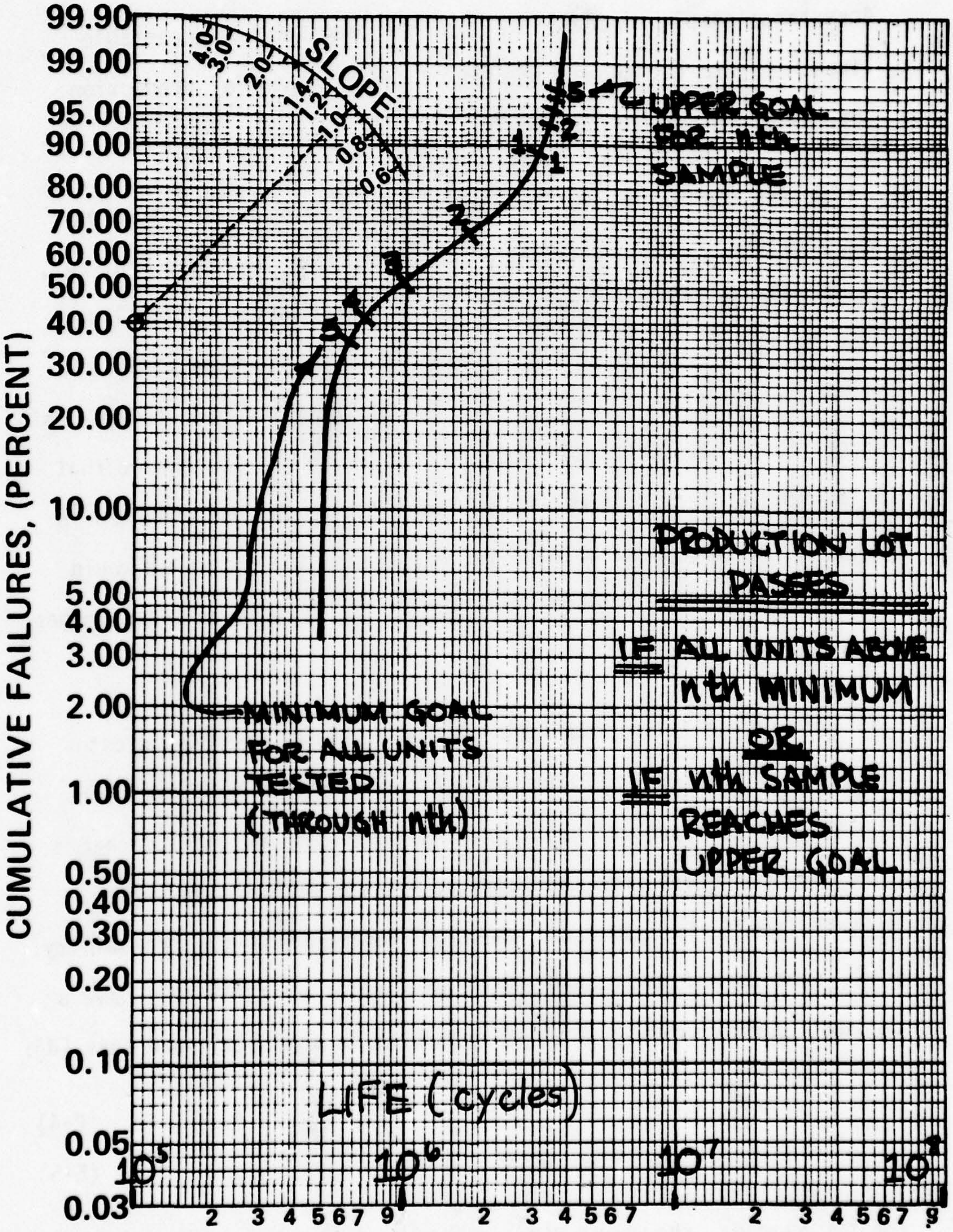

Fig. E-7. Graphical Presentation of Decision Table.

3,375,000 cycles and a 94.9% chance of the second unit failing before 3,375,000 cycles. Therefore, the probability of a second pump in sequential sampling failing before 3,375,000 cycles is:

$$(94.9\%)\ (94.9\%) \ = \ 90.0\% \qquad \text{(E-6)}$$

Therefore, if the distribution is barely acceptable, there is less than a 10% chance of getting more than 3,375,000 cycles during the second test. Following the same reasoning for other sample sizes, we see that the upper goal of the nth pump is:

$$F(x)^n \ = \ 0.9 \qquad \text{(E-7)}$$

or $$F(x) \ = \ (0.9)^{1/n} \qquad \text{(E-8)}$$

which is reflected in Table E-2.

The second question we must ask is: "If both samples are from the acceptable distribution or a better distribution, what minimum life do we expect them to each attain?" The equation for our question is:

$$(1 - F(x))\ (1 - F(x)) \ = \ 0.1 \qquad \text{(E-9)}$$

or $$(1 - F(x))^2 \ = \ 0.1 \qquad \text{(E-10)}$$

or $$F(x) \ = \ 1 - (0.1)^{\frac{1}{2}} \qquad \text{(E-11)}$$

or $$F(x) \ = \ 0.684 \text{ or } 68.4\% \qquad \text{(E-12)}$$

The life associated with 68.4% cumulative failures is 1,750,000 cycles. Thus, we expect 68.4% of the units to fail before 1,750,000 cycles. Or, only 31.6% of the units will last longer than 1,750,000 cycles. Thus, the probability that we will get two units in sequence that last

TABLE E-2. Decision Table. Sample Life Goals for Acceptance of Production Lot of Pumps. Consumer's Risk is Less Than or Equal to 10%.

SAMPLE NO. TESTED	MINIMUM LIFE FOR ALL TESTED $F(X) = 1-(0.1)^{1/n}$		GOAL FOR TESTED SAMPLE $F(X) = (0.9)^{1/n}$	
	F(X)	LIFE	F(X)	LIFE
1	90.0%	3,125,000	90.0%	3,125,000
2	68.4%	1,750,000	94.9%	3,375,000
3	53.6%	1,000,000	96.5%	3,540,000
4	43.8%	710,000	97.4%	3,625,000
5	36.9%	625,000	97.9%	3,665,000

NOTE: IF nTH SAMPLE MEETS UPPER GOAL OR ALL TESTED UNITS MEET MINIMUM LIFE REQUIREMENT FOR nTH SAMPLE, LOT IS ACCEPTED.

longer than 1,750,000 cycles is:

$$(31.6\%)\ (31.6\%) = 10\% \tag{E-13}$$

which is our acceptable level of risk.

Extending this procedure to other sample sizes, we find that the minimum life for all samples through the nth is:

$$(1 - F(x))^n = 0.1 \tag{E-14}$$

or $$F(x) = 1 - (0.1)^{1/n} \tag{E-15}$$

The results of the test plan are presented in the Decision Table, Table E-2.

EXAMPLE

To illustrate how the procedure works, consider Data Set 8 (Appendix E-C). The pumps used to obtain Data Set 8 are allegedly of the same design as those of Data Set 7. Therefore, we will accept the production lot if they pass the test outlined in Table E-2.

The first pump failed at 2,576,000 cycles. Since this figure is less than 3,125,000, another unit must be tested. The second pump failed at 1,240,000 cycles. Since this figure is neither greater than 3,375,000 cycles nor greater than 1,750,000, another pump is tested.

The third pump lasted longer than 1,000,000 cycles. Since the other pumps also lasted longer than 1,000,000 cycles, the test could have been terminated after 1,000,000 cycles and the lot accepted.

The fact that the pump lasted more than the 3,540,000 cycle upper goal would have also allowed our accepting the lot.

Data Set 8 represents the results of a production lot tested using an available industrial acceptance procedure. Compared with the procedure used to accept the lot represented by Data Set 8, the procedure outlined in Table E-2 could have saved 2,850,000 test cycles. This represents a potential savings of 37% of the total test time. In other words, the acceptance procedure used with Data Set 8 took 59% longer to run than the proposed procedure.

CLOSURE

The procedure outlined in this paper requires the manufacturer to demonstrate that a production lot is acceptable based on a desired distribution. Other procedures which require that the manufacturer demonstrate the subsequent production lots have the same distribution as the Design Acceptance sample increase the producer's risk and the cost of testing.

Should the production lot be "weaker" than the original lot, the distribution "width" decreases and both the characteristic life and the minimum life decrease. These "decreases" both act to reduce the chances of the "lot" passing the test.

It should be obvious that the producer controls the producer's risk by making the product "better" than required. The procedure of requiring production lots to only be better than initially desired helps to keep the producer's risk at a reasonable level and save

production costs.

If the producer desires to run more than five units to demonstrate the acceptability of a lot, the Decision Table can easily be extended. This allows the producer to make decisions regarding cost "trade-offs" between testing and improved designs to lower the producer's risk.

Procedures of the type outlined in Table E-2 also provide the consumer with higher reliability if the producer's risk is decreased. Thus, the producer who "improves" a product actually reduces the producer risk associated with the sampling procedure while increasing the component reliability.

ACKNOWLEDGEMENTS

The authors wish to extend their sincere appreciation to the U.S. Army MERADCOM, the Naval Ship Engineering Center, the Department of Energy, and the sponsors of the Basic Fluid Power Research Program for providing the support, motivation, and guidance which made this paper possible. Special thanks are extended to those members of the BFPR who shared test data which were used for the development of the procedure presented in this paper.

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3. Heller, R. A., Editor, Probabilistic Aspects of Fatigue, STP 511, American Society for Testing and Materials, p. 12, p. 63.

4. Miller, Irwin, and J. E. Freund, Probability and Statistics for Engineers, Prentice-Hall, Inc., Englewood Cliffs, New Jersey, 1965.

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APPENDIX E-A

A DESIGN MECHANICS TECHNICAL NOTE

ACCEPTANCE SAMPLING RISKS

There are two parties interested in acceptance sampling—the producer and the consumer. The producer is interested because acceptance sampling carries a risk of rejecting "good" lots. The consumer is interested because acceptance sampling carries a risk of accepting "bad" lots. These risks are defined as follows:

Producer's Risk—The Producer's Risk (α) is the probability that a "good" lot will be rejected by the sampling plan.

Consumer's Risk—The Consumer's Risk (β) is the probability that a "bad" lot will be accepted by the sampling plan.

Typical values for producer and consumer risks range from 0.01 to 0.10. For some sampling plans, the risk values are fixed. In some sampling plans, the risk values are a variable.

The use of the words "good" and "bad," as they relate to acceptance sampling, have more specific meanings:

"Good" Lot—A lot where the maximum percent defective is less than or equal to some specified value.

"Bad" Lot—A lot where the percent defective is greater than some specified value.

Figure 1 shows the performance of an ideal sampling plan. There are no risks.

Figure 2 shows how a typical sampling plan might really perform. In this example, for a given percent defective, there is a finite

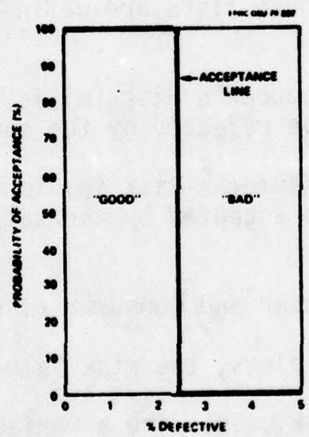


Fig. 1. Ideal Sample Plan Performance: Zero Percent Chance of Accepting a Bad Lot and 100 Percent Chance of Accepting a Good Lot.

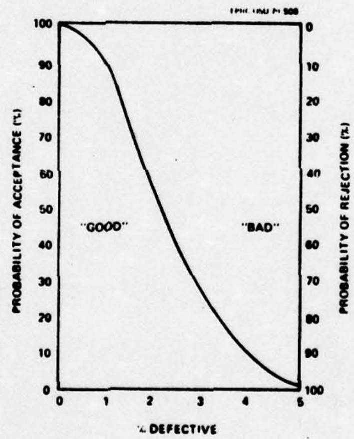


Fig. 2. Typical Sample Plan Performance: 90% Chance of Accepting 1% Defective Lot and 90% chance of Rejecting 4% Defective.

probability of acceptance. Naturally, there is also a finite probability of rejection.

Figure 3 illustrates how the producer defined "good" lot and the consumer defined "bad" lot related to one other. Compared with the "ideal plan," the "realistic" example of Fig. 3 shows that the following must occur:

1. The producer must assume some risk.
2. The consumer must assume some risk.

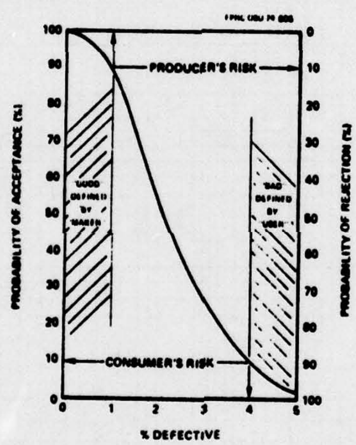
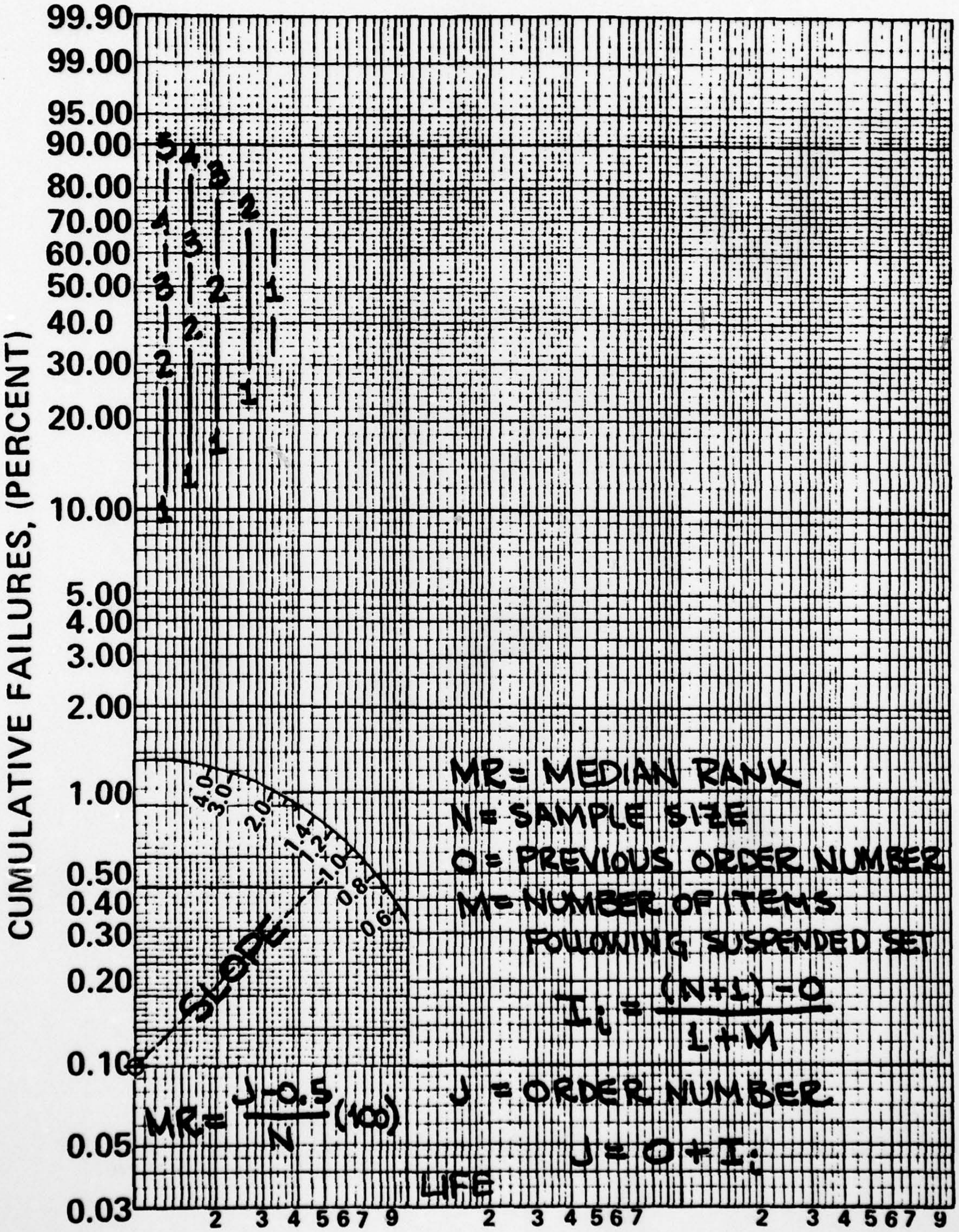


Fig. 3. Example Sample Plan Performance: 10% Consumer's Risk for 4% Defective and 10% Producer's Risk for 1% Defective.

APPENDIX E-B

WEIBULL FAILURE ANALYSIS CHART



APPENDIX E-C

DATA SET 7

Component	Pump
Material	Aluminum
Identification	780810 – Run 1
Temperature	82°C (180°F)
Pressure	1.2 Rated
Cycle	6–10 cps, 50-50
Failure Mode	Fatigue

UNIT	CYCLES	CONDITION
1	1,305,000	Failure
2	5,183,000	Suspension
3	4,413,000	Failure
4	2,089,000	Failure
5	1,206,000	Failure

DATA SET 8

Component	Pump
Material	Aluminum
Identification	780810 – Run 2
Temperature	82°C (180°F)
Pressure	1.2 Rated
Cycle	6–10 cps, 50-50
Failure Mode	Fatigue

UNIT	CYCLES	CONDITION
1	2,576,000	Failure
2	1,240,000	Failure
3	3,850,000	Failure

APPENDIX F

COMPUTER-AIDED WEIBULL ANALYSIS OF FAILURE DATA

The Weibull distribution is one of the most popular statistical methods for predicting component life. Estimates for percent survival at a specified life for a production lot may be obtained by testing a smaller number of representative samples for that lot. The Weibull distribution is popular not only because of its goodness-of-fit to most component failure data but also for its ease of implementation by graphical techniques. The capability to analyze data quickly "by hand" is a great aid in interpretation and understanding. A variety of technical publications and statistics texts cover the theory and application of the Weibull distribution for life estimation. A few of these publications are included in the references at the end of this appendix. Rather than discussing the statistical basis for the Weibull distribution or presenting in great detail the common procedure for performing analysis by hand, this paper presents a computer-aided method of analyzing data with the added capabilities of handling non-zero minimum life and data including samples upon which testing has been suspended prior to failure.

Plotting Failure Data Using the Weibull Distribution

The two-parameter Weibull distribution may be expressed in the form

$$F = 1 - e^{-(x/\theta)^{\beta}} \qquad \text{(F-1)}$$

where: x = random variable (cycles to failure)

F = cumulative distribution (percent failed in the sample group ÷ 100)

θ = characteristic life

β = shape parameter or Weibull slope

Manipulation of this expression gives

$$\ell n\left(\ell n\left(\frac{1}{1-F}\right)\right) = \beta\ \ell n x - \beta\ \ell n\theta$$

which is of the linear form $y = mx' + b$.

where: $y = \ell n\left(\ell n\left(\frac{1}{1-F}\right)\right)$

$m = \beta$

$x' = \ell n\ x$

$b = -\beta\ \ell n$

It can be seen that proper scaling of the axes will enable the plot of Eq. (F-1) to be a straight line. Weibull graph paper having this scaling (See Fig. F-1.) simplifies the analysis to the following steps:

1. Determine and assign a value of median rank to each failed item in the sample group.
2. Plot each failure on Weibull probability paper. (Median rank and life-to-failure are the ordinate and abscissa, respectively.)
3. Draw the best-fit straight line through the data points.
4. Determine β and θ.

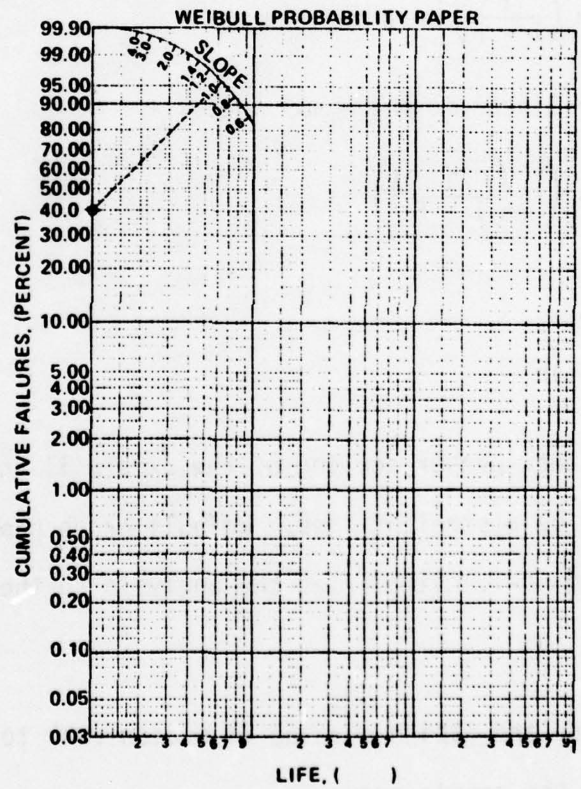


Fig. F-1. Weibull Probability Paper.

Plotting Failure Data with Suspended Items

Sometimes life testing is not completed to failure for every item in a sample group. The components upon which testing was ended prematurely are referred to as suspended items. The information obtained from these items is useful in analysis even though failure did not occur. Incorporating suspended item data into the failure data may be accomplished by using the procedure in Appendix F-A, which modifies the value of median rank assigned to each failed item. Although this procedure appears complicated at first glance, analysis "by hand" is still quite easy.

Plotting Failure Data with Non-Zero Minimum Life

Many times in component life testing there appears to be a value for life below which no failures occur. Data, which have such a cumulative life distribution exhibiting non-zero minimum life, result in a two-parameter Weibull plot which is convex, as shown in Fig. F-2. Note that the heavy line in this plot is simply a best-fit second order curve which does not necessarily conform to the Weibull distribution. That is, there are not values of characteristic life θ and shape parameter β which can be used with Eq. (F-1) to obtain the curve shown, since, as discussed earlier, all plots of the function of Eq. (F-1) will be straight lines on Weibull paper.

To avoid this problem, a three-parameter Weibull distribution may be used:

$$F = a - e^{-\left(\frac{x - \delta}{\theta - \delta}\right)^{\beta}} \qquad \text{(F-2)}$$

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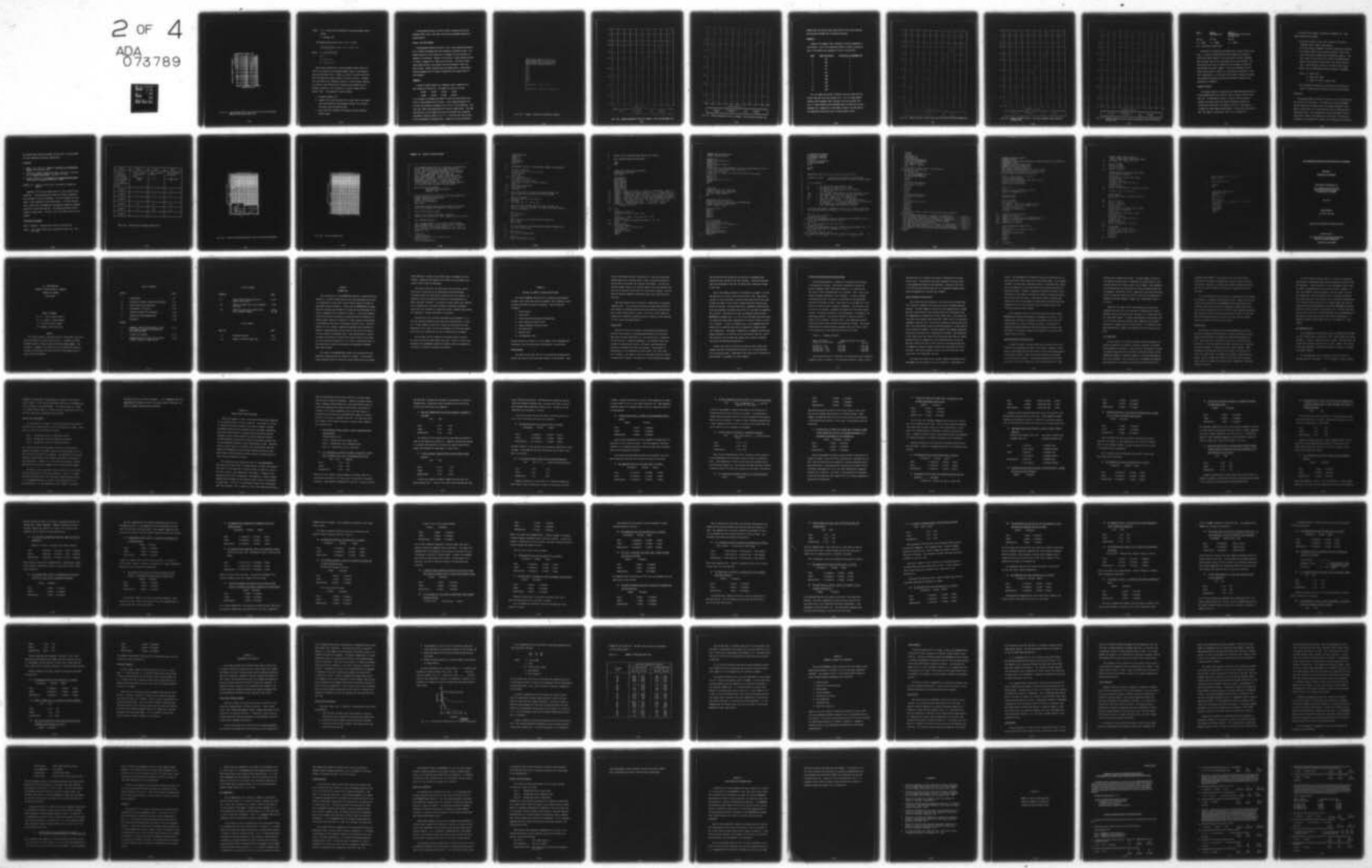
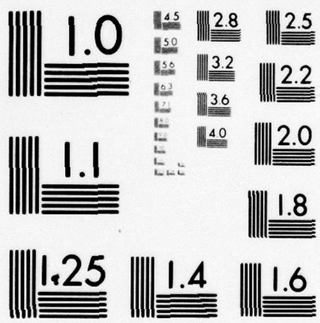




MICROCOPY RESOLUTION TEST CHART

NATIONAL BUREAU

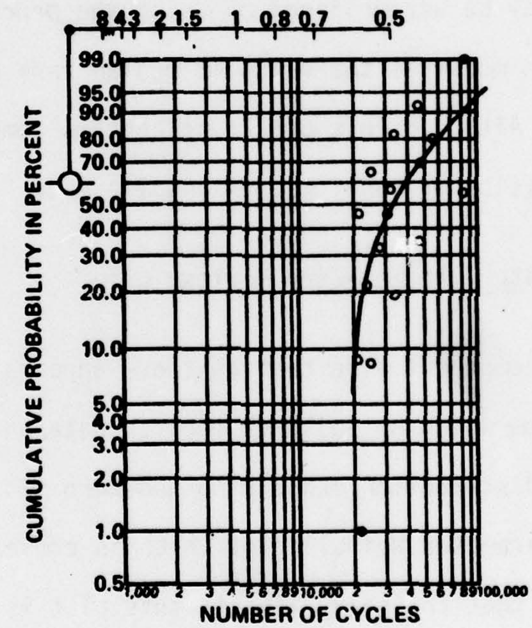


Fig. F-2. Two-Parameter Weibull Distribution Plot of Failure Data Showing a Non-Zero Minimum Life.

where: F, X, θ, and β are as defined for the two-parameter distribution

δ = minimum life

The equation may be put in the y = mx' + b form:

$$\ell n \left(\ell n \left(\frac{1}{1 - F} \right) \right) = \beta \, \ell n \, (x - \delta) - \beta \, \ell n \, (\theta - \delta)$$

where:
$$y = \ell n \left(\ell n \left(\frac{1}{1 - F} \right) \right)$$

$$m = \beta$$

$$x' = \ell n \, (x - \delta)$$

$$b = -\beta \, \ell n \, (\theta - \delta)$$

Hence, data conforming to a three-parameter Weibull model will plot as a straight line on standard Weibull paper if the parameter along the horizontal axis is number of cycles to failure minus minimum life rather than simply number of cycles to failure. Although this distribution will generally result in a more accurate, meaningful analysis, the difficulties of obtaining the optimum value of minimum life make it less attractive as a quick, simple failure analysis tool. The sequence of steps involved:

1. Estimate minimum life.
2. Subtract this value from the life of each item in the sample group and perform the two-parameter analysis as discussed previously upon the modified data.
3. Draw a best-fit straight line through the data plotted on Weibull paper.

If being done by hand, an initial value of minimum life may be obtained from a plot of the data using the two-parameter method discussed earlier.

WEIBULL PLOTTING PROGRAM

Three-parameter Weibull plotting is not a very complicated process, but is usually considered too time-consuming to perform by hand. The answer obviously is the creation of a program to do the analysis on computer or calculator. Appendix F-B contains a simple program written in Fortran language for a time-sharing terminal. With minor input/output modifications, the program could be performed in batch job mode instead. Before examining how the program works, a few Weibull plotting examples will be shown to demonstrate the capabilities of this program.

<u>Example 1</u>

A group of eight samples of a component from a production lot were tested to failure [1]. The number of cycles at failure:

22,000	25,000	30,000	33,000
35,000	52,000	63,000	104,000

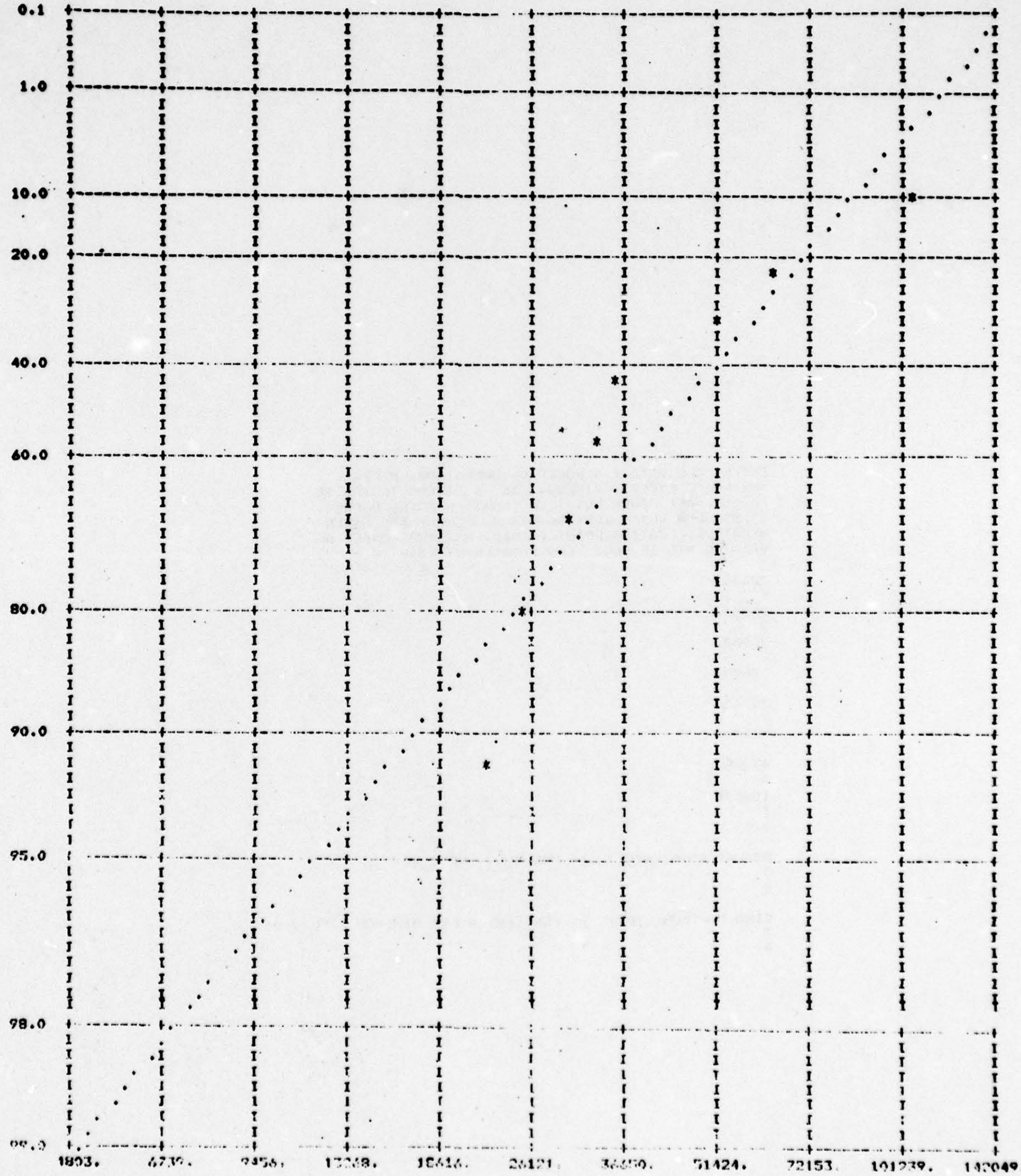
It is desired to estimate the number of cycles for which 90% of the units in the production will survive. First assuming minimum life of zero, the printout and Weibull plot of Fig. F-3 are obtained. From this plot, 90% of the population will survive 17,000 cycles. The same data were run in the program with non-zero minimum life, producing the Weibull analysis shown in Fig. F-4. From this plot, 90% survivability corresponds to 22,500 cycles. Comparison of the correlation

```
ENTER LIFE DATA (IN HOURS OF OPERATION, MILES,
NUMBER OF CYCLES, STRESS, ETC. )  ITEMS TESTED TO
FAILURE ARE GIVEN POSITIVE FLOATING POINT VALUES.
ITEMS UPON WHICH TESTING WAS SUSPENDED ARE GIVEN
NEGATIVE FLOATING POINT VALUES. MAXIMUM NUMBER OF
DATA POINTS IS 100.  END DATA ENTRY WITH  0.0    .
?
22000.
?
25000.
?
30000.
?
33000.
?
35000.
?
52000.
?
63000.
?
104000.
?
0.0

REPEAT DATA ENTRY?  (1 FOR YES, 0 FOR NO)
?
0

FIND MINIMUM LIFE?  (1 FOR YES, 0 FOR MINIMUM LIFE=0.0)
?
0
```

Fig. F-3a. Example 1 Data Entry for Weibull Program.



RELIABILITY VERSUS LIFE

MINIMUM LIFE	CHARACTERISTIC LIFE	BETA	CORR. COEF.
0	52155.	[illegible]	[illegible]

Fig. F-3b. Output and Weibull Plot for example 1 with zero minimum life.

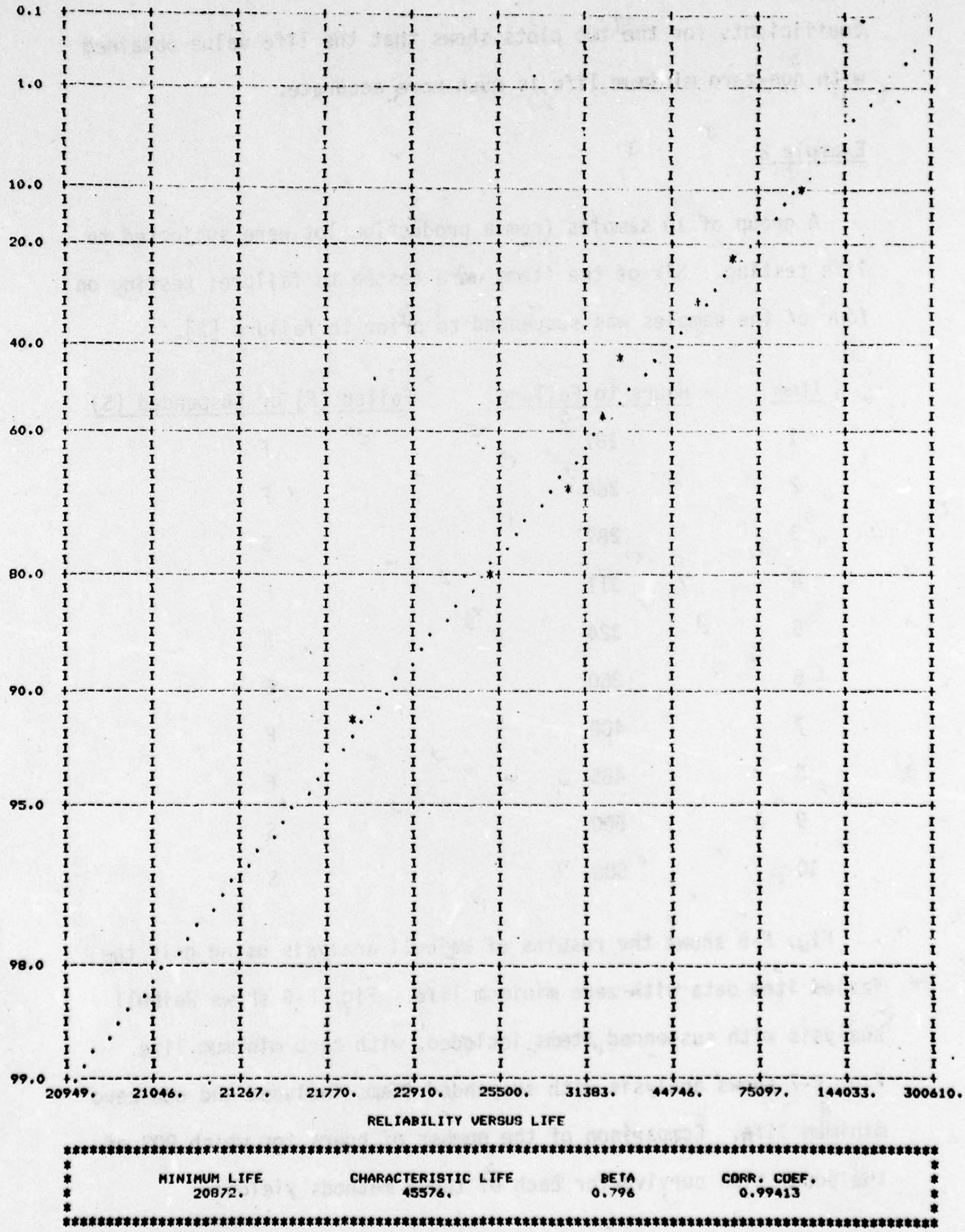


Fig. F-4. Output and Weibull Plot for example 1 with non-zero minimum life.

coefficients for the two plots shows that the life value obtained with non-zero minimum life is much more accurate.

Example 2

A group of 10 samples from a production lot were subjected to life testing. Six of the items were tested to failure; testing on four of the samples was suspended to prior to failure [2].

Item	Hours to Failure	Failed (F) or Suspended (S)
1	181	F
2	268	F
3	287	S
4	311	F
5	324	S
6	360	F
7	408	F
8	485	F
9	500	S
10	500	S

Fig. F-5 shows the results of Weibull analysis using only the failed item data with zero minimum life. Fig. F-6 shows Weibull analysis with suspended items included, with zero minimum life. Fig. F-7 shows analysis with suspended items included and non-zero minimum life. Comparison of the number of hours for which 90% of the population survive for each of those methods yields:

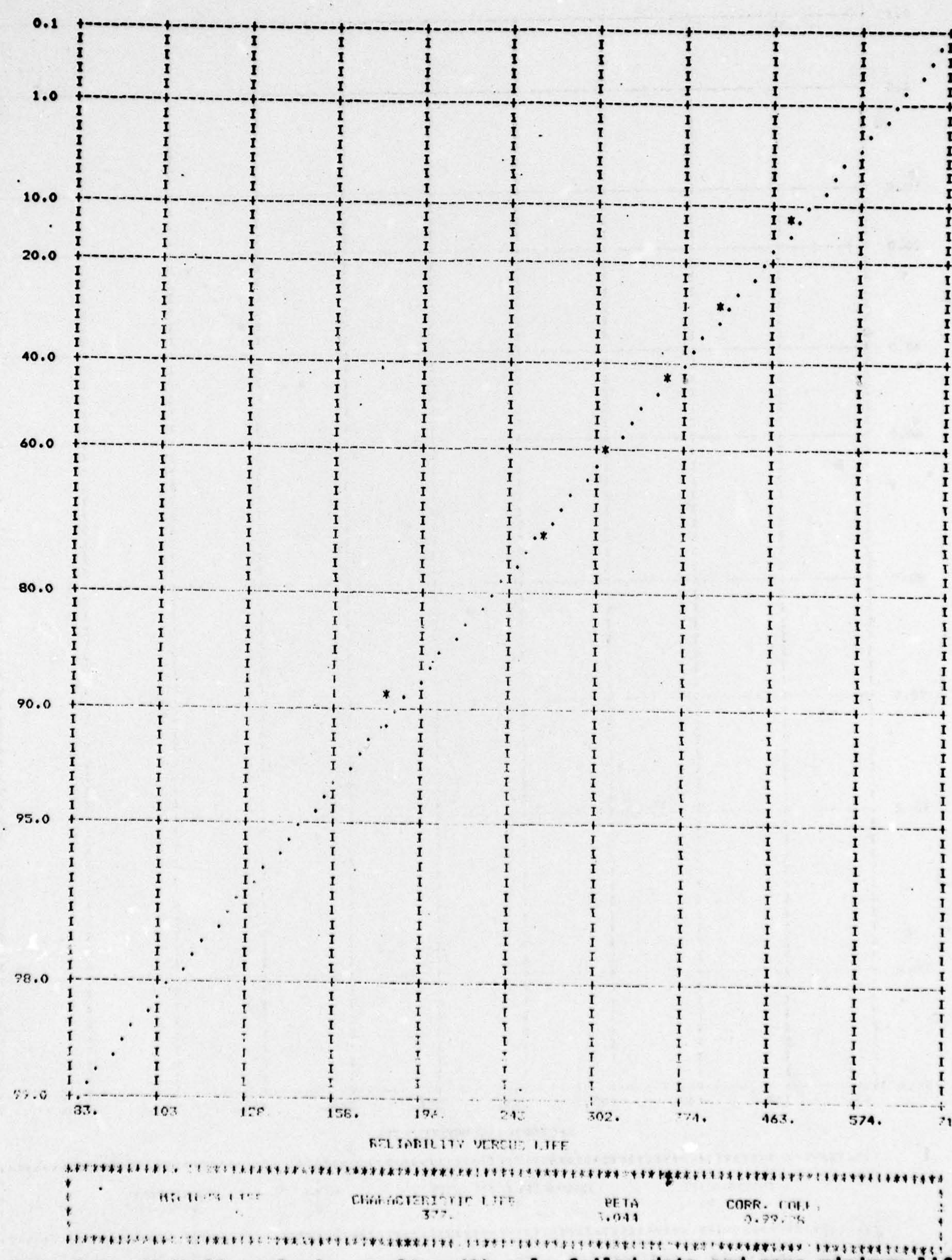
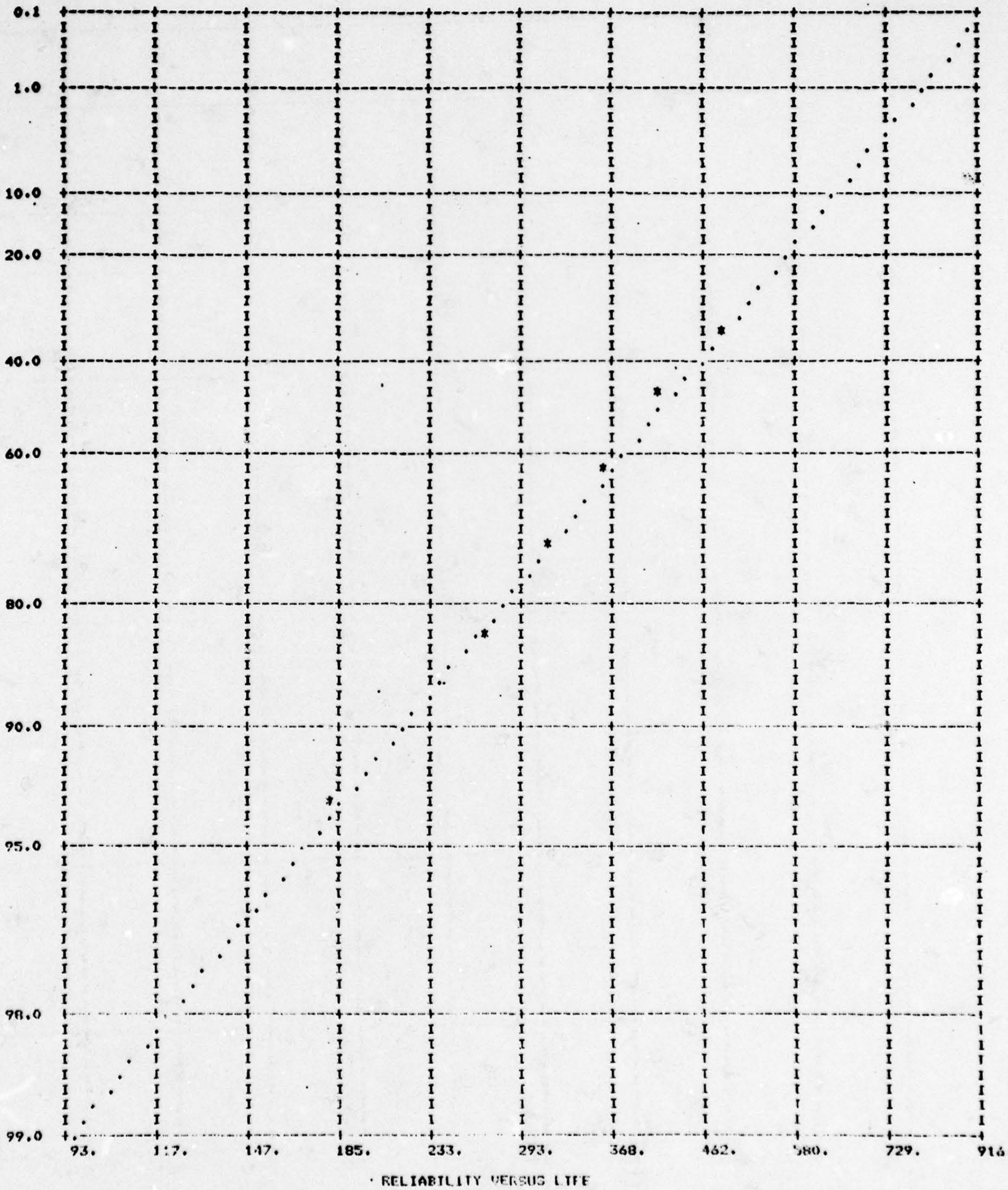
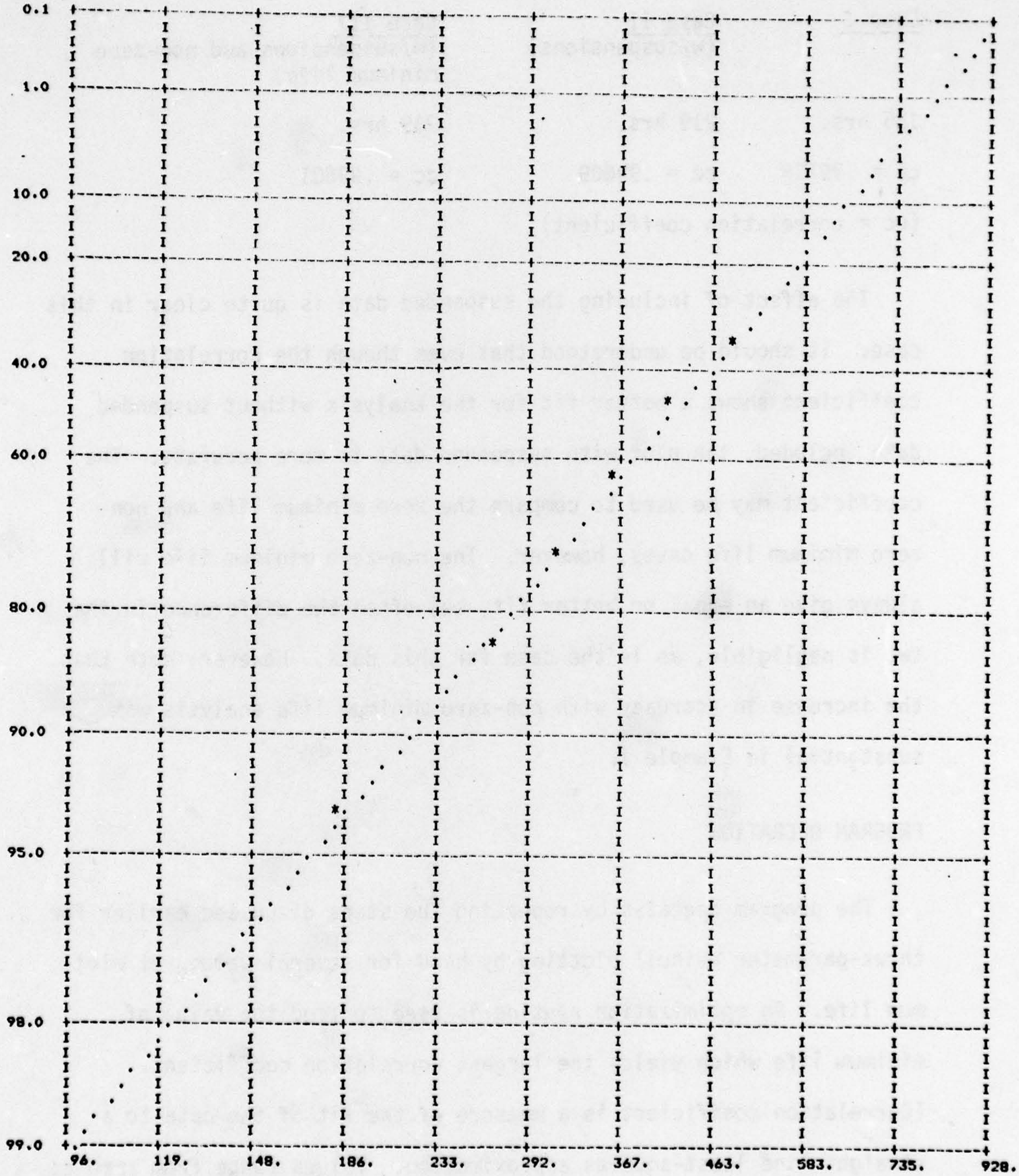


Fig. F-5. Weibull analysis results with only failed data and zero minimum life,



MINIMUM LIFE	CHARACTERISTIC LIFE	BETA	CORR. COEF.
0.	[illegible]	2.561	[illegible]

Fig. F-6. Weibull analysis results including suspended items with zero minimum life.



RELIABILITY VERSUS LIFE

MINIMUM LIFE	CHARACTERISTIC LIFE	BETA	CORR. COEF.
8.	467.	2.780	0.99601

Fig. F-7. Weibull analysis results including suspended items with non-zero minimum life.

Case I	Case II (w/suspensions)	Case III (w/suspensions and non-zero minimum life)
185 hrs.	219 hrs.	219 hrs.
cc = .99728	cc = .99609	cc = .99601

(cc = correlation coefficient)

The effect of including the suspended data is quite clear in this case. It should be understood that even though the correlation coefficient shows a better fit for the analysis without suspended data included, the plot with suspended data is more accurate. The coefficient may be used to compare the zero minimum life and non-zero minimum life cases, however. The non-zero minimum life will always give an equal or better fit, but often the difference in the two is negligible, as is the case for this data. However, note that the increase in accruacy with non-zero minimum life analysis was substantial in Example 1.

PROGRAM OPERATION

The program operates by repeating the steps discussed earlier for three-parameter Weibull plotting by hand for several values of minimum life. An optimization routine is used to find the value of minimum life which yields the largest correlation coefficient. (Correlation coefficient is a measure of the fit of the data to a straight line least-squares approximation. Values range from zero to one. Zero means no correlation, and 1.0 is a perfect fit.)

A listing of the program is contained in Appendix F-B. Some notes concerning the program:

* Data may be entered in any order; ordering of the data in ascending value is done in the program.
* Up to 100 values of component life may be entered and analyzed.
* The program contains its own plotting routine. Axes of the plots are scaled as common Weibull probability paper is. However, labels on the vertical axis differ from common Weibull paper in that percent survived rather than percent failed is used. (Ninety percent survived is the same as 10% failed.)
* Median rank is approximated using the formula $M = \frac{J - 0.3}{N + 0.4}$.

  where: M = median rank

  J = mean order number

  N = number of items in sample group
* More information concerning the optimization routine may be found in reference [3], from which the routine was obtained.

CONCLUSION

Two-parameter Weibull plotting has proven to be an invaluable tool for analysis of failure data. For those who use the two-parameter Weibull plotting on a regular basis, the program contained in this paper is quite useful as a time-saving tool and for checking plots of data done by hand. The incorporation of non-zero minimum life will enable more accurate life predictions in many cases. The decision of whether to incorporate this feature can be made quickly and easily

by analyzing with both zero minimum life and with non-zero minimum life then comparing correlation coefficients.

REFERENCES

1. Kapur, K. C., and L. R. Lamberson, Reliability in Engineering Design, Wiley and Sons, 1977.

2. "Plotting on Weibull Probability Paper," Module 12 in the Ford Series, Reliability Methods, Ford Motor Co., 1972.

3. Mischke, Charles R., An Introduction to Computer-Aided Design, Prentice-Hall, Englewood Cliffs, N.J., 1968.

APPENDIX F-A. Plotting Failure Data in the Weibull Probability Domain.

Component life can be estimated based on a few failures from a population. Tested specimens which have not failed, suspensions, add confidence to the life estimates. The life estimator for this "Note" is the cumulative failure distribution. A Failure Analysis Chart, shown in Fig. F-A1, summarizes the equations needed for graphing cumulative failure distributions. The graphing procedure with an example is shown below. The plot in Fig. F-A2 shows a plot for the example.

Procedure with Example

Steps 1 through 4: Proceed across chart for each failure.

Step 5: Plot "Locks" ranks versus respective failure life. See Fig. F-A2.

STEP 1	STEP 2	STEP 3	STEP 4
List the data in ascending order. Include suspended items. Code failure (F) and suspensions (S). Exclude suspensions before first failure.	Calculate "Increments" for each group of failures: $I = \frac{(N+1)-P}{1+F}$	Calculate mean order number, J: $J = P + I$	Calculate "Locks" rank, M, for failures: $M = \frac{J - 0.5}{N}(100)$
1,305,000 (F)	1	1	8%
1,370,000 (F)	1	2	25%
2,159,000 (S)	–	–	–
2,303,000 (F)	1.25	3.25	46%
4,413,000 (F)	1.25	4.50	67%
5,183,000 (S)	–	–	

Table F-A1. Calculations for Example Weibull Plot.

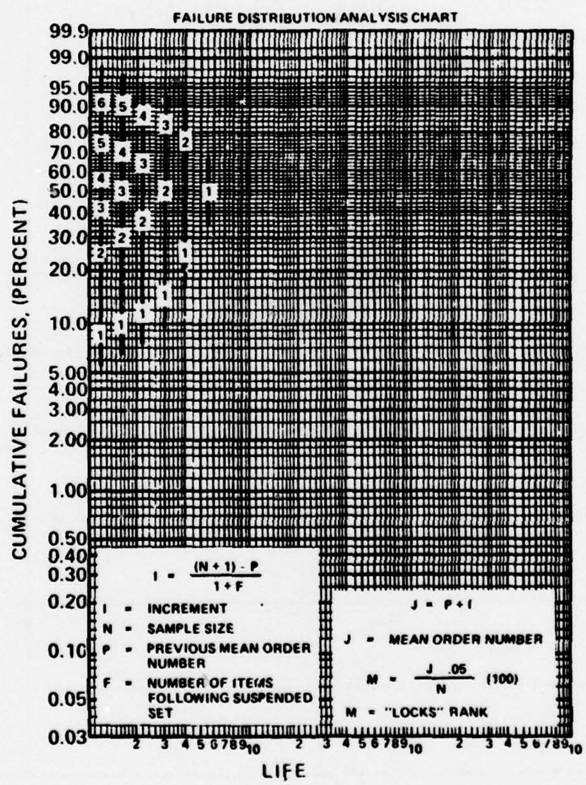


Fig. F-A1. Failure Distribution Analysis Chart with Plotting Equations.

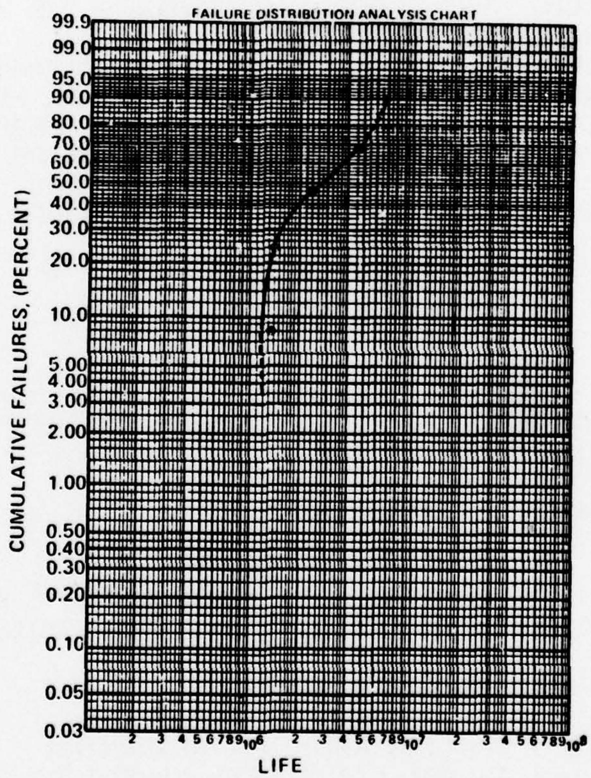


Fig. F-A2. Plot of Example Data.

APPENDIX F-B. **Weibull Plotting Program.**

```
C-------------------------------------------------------------------
C
C     THIS PROGRAM MAKES USE OF THE WEIBULL DISTRIBUTION
C     TO ANALYZE AND DISPLAY DATA FROM RELIABILITY
C     TESTS.  BOTH FAILED AND SUSPENDED TEST DATA MAY
C     BE ENTERED.  THE PROGRAM DETERMINES THE MINIMUM
C     LIFE, THE CHARACTERISTIC LIFE, AND THE
C     SHAPE PARAMETER BETA.  THESE THREE PARAMETERS
C     DETERMINE A STRAIGHT LINE ON A PLOT OF PER CENT
C     CUMULATIVE FAILURE VS. LIFE WITH THE PROPER
C     TRANSFORMATION OF AXES.  A COMPARISON OF THE
C     FITTED CURVE AND THE TEST DATA IS ACCOMPLISHED BY
C     PLOTTING THE TEST DATA ON THE SAME GRAPH.
C
C     WRITTEN BY DICK HEADRICK
C                OKLAHOMA STATE UNIVERSITY
C                JANUARY 1979
C
C-------------------------------------------------------------------
      DIMENSION DLIFE(100),POS(100),FAIL(100),RANK(100)
      COMMON /SENSE/FAIL,RANK,NFAIL
      COMMON /TWO/A1,A2,A3
      DATA DLIFE,POS/200*0.0/
      DATA NW/6/
C
C     SUBROUTINE ENTER READS IN THE VECTOR OF LIFE DATA
C     DLIFE AND RETURNS THE NUMBER OF DATA POINTS N
C
      CALL ENTER(DLIFE,N,IFIND)
C
C     SORTEM IS A SORTING ROUTINE, RETURNING
C     LOWEST LIFE AS DLIFE(1), NEXT LOWEST AS DLIFE(2),ETC
C
      CALL SORTEM(DLIFE,N)
C
C     THE FOLLOWING GROUP OF STATEMENTS ASSIGNS MEAN
C     ORDER NUMBERS(POS) TO THE FAILED DATA.  SUSPENDED
C     DATA RECEIVE A MEAN ORDER NUMBER OF 0.0.  THE
C     FIRST FAILURE IS FOUND AND ASSIGNED THE MEAN ORDER
C     NUMBER OF 1.0.
C
      IFAIL1=0
100   IFAIL1=IFAIL1+1
      IF(DLIFE(IFAIL1).LT.0.0) GO TO 100
      POS(IFAIL1)=1.0
      FAIL(1)=DLIFE(IFAIL1)
```

```
      IFAIL=IFAIL1+1
      ISUSP=0
      AINC=1.0
      PREV=1.0
      AN=FLOAT(N)
      NFAIL=1
C
C     SUBSEQUENT VALUES OF MEAN ORDER NUMBER ARE ASSIGNED
C
      DO 300 I=IFAIL,N
      IF(DLIFE(I).GT.0.0)  GO TO 200
      ISUSP=ISUSP+1
      GO TO 300
200   NFAIL=NFAIL+1
      FAIL(NFAIL)=DLIFE(I)
      IF(ISUSP.EQ.0) GO TO 400
      AINC=(AN+1.0-PREV)/(AN+2.0-FLOAT(I))
      ISUSP=0
400   POS(NFAIL)=PREV+AINC
      PREV=POS(NFAIL)
300   CONTINUE
C
C     FROM THE VALUES OF POSITION ASSIGNED ABOVE THE
C     MEDIAN RANK IS CALCULATED FOR EACH FAILURE
C
      DO 500 I=1,NFAIL
      RANK(I)=(POS(I)-0.3)/(AN+0.4)
500   CONTINUE
      IF(IFIND .EQ. 0)  GO TO 600
C
C     OPTIMIZATION ROUTINE GOLD1 IS USED TO FIND THE
C     BEST-FIT VALUE OF MINIMUM LIFE WHEN NON-ZERO MINIMUM
C     LIFE IS SPECIFIED.
C
      XL=0.0
      XR=0.99*FAIL(1)
      F=0.0001
      K=0
      CALL GOLD1 (XL,XR,F,YBIG,XBIG,XLF,XRF,K,N)
      AMINLF=XBIG
      GO TO 700
C
C     THE FOLLOWING IS PERFORMED INSTEAD WHEN MINIMUM LIFE
C     OF ZERO IS SPECIFIED.
C
600   AMINLF=0.0
      CALL MERIT1(AMINLF,DUM)
C
700   BETA=A2
      CHARLF=AMINLF+EXP(-A1/A2)
```

```
C
C     WEIPLT PLOTS THE DATA AND PRINTS THE OUTPUT.
C
      CALL WEIPLT(AMINLF,BETA,CHARLF)
C
      STOP
      END
C
C
C
      SUBROUTINE ENTER(DLIFE,N,IFIND)
      DIMENSION DLIFE(100)
      DATA NW/6/
C
      WRITE(NW,5)
      WRITE(NW,10)
      WRITE(NW,20)
      WRITE(NW,30)
      WRITE(NW,40)
      WRITE(NW,50)
      WRITE(NW,60)
5     FORMAT(////)
10    FORMAT(' ENTER LIFE DATA (IN HOURS OF OPERATION, MILES,')
20    FORMAT(' NUMBER OF CYCLES, STRESS, ETC. )  ITEMS TESTED TO')
30    FORMAT(' FAILURE ARE GIVEN POSITIVE FLOATING POINT VALUES.')
40    FORMAT(' ITEMS UPON WHICH TESTING WAS SUSPENDED ARE GIVEN')
50    FORMAT(' NEGATIVE FLOATING POINT VALUES. MAXIMUM NUMBER OF')
60    FORMAT(' DATA POINTS IS 100.  END DATA ENTRY WITH  0.0    .')
80    FORMAT(/,' REPEAT DATA ENTRY?  (1 FOR YES, 0 FOR NO)')
90    FORMAT(/,' FIND MINIMUM LIFE?  (1 FOR YES, 0 FOR',
     *  ' MINIMUM LIFE=0.0)')
C
300   I=0
310   I=I+1
      READ(5,*) DLIFE(I)
      IF(DLIFE(I).NE.0.0) GO TO 310
      N=I-1
      IF(N.GT. 0   .AND.  N.LE.100) GO TO 320
      WRITE(NW,70) N
70    FORMAT(' ERROR IN SUBROUTINE ENTER   N=',I5)
      STOP
320   WRITE(NW,80)
      READ(5,*) IREPET
      IF(IREPET .EQ. 1) GO TO 300
      WRITE(NW,90)
      READ(5,*) IFIND
      RETURN
      END
C
```

```
C
      SUBROUTINE OUTPUT(A,B,C)
      COMMON /TWO/A1,A2,A3
C
      WRITE(6,5)
5     FORMAT(/,15X,101('*'))
      WRITE(6,6)
6     FORMAT(15X,'*',99X,'*')
      WRITE(6,7)
7     FORMAT(15X,'*',10X,'MINIMUM LIFE',10X,'CHARACTERISTIC LIFE',
     * 10X,'BETA',10X,'CORR. COEF.',13X,'*')
      WRITE(6,8) A,C,B,A3
8     FORMAT(15X,'*',10X,F10.0,14X,F10.0,14X,F7.3,11X,F8.5,
     * 15X,'*')
      WRITE(6,6)
      WRITE(6,9)
9     FORMAT(15X,101('*'),////)
      RETURN
      END
C
C
      BLOCK DATA
      DIMENSION FAIL(100),RANK(100)
      COMMON /SENSE/FAIL,RANK,NFAIL
      DATA FAIL,RANK/200*0.0/
      END
C
C
      SUBROUTINE MERIT1(DEL,CC)
      DIMENSION X(100),Y(100),FAIL(100),RANK(100)
      COMMON /SENSE/FAIL,RANK,NFAIL
      COMMON /TWO/A1,A2,A3
C
      AN=FLOAT(NFAIL)
      SUMX=0.
      SUMX2=0.
      SUMY=0.
      SUMXY=0.
      SUMY2=0.
C
      DO 10 I=1,NFAIL
      X(I)=ALOG(FAIL(I)-DEL)
      Y(I)=ALOG(ALOG(1./(1.-RANK(I))))
      SUMX=SUMX+X(I)
      SUMX2=SUMX2+X(I)*X(I)
      SUMY=SUMY+Y(I)
      SUMXY=SUMXY+X(I)*Y(I)
      SUMY2=SUMY2+Y(I)*Y(I)
10    CONTINUE
C
```

```
      R1=AN*SUMXY-SUMX*SUMY
      R2=AN*SUMX2-SUMX**2
      R3=AN*SUMY2-SUMY**2
      A2=R1/R2
      A1=SUMY/AN-A2*SUMX/AN
      CC=R1/SQRT(R2*R3)
      A3=CC
C
      RETURN
      END
C
C
      SUBROUTINE GOLD1 (XL,XR,F,YBIG,XBIG,XLF,XRF,K,N)
C
C     SUBROUTINE GOLD1 -- A ROUTINE TO MAXIMIZE A FUNCTION OF
C                         ONE VARIABLE USING A GOLDEN SECTION SEARCH.
C
C     KEY TO SYMBOLS --
C
C     XL        --  THE SPECIFIED LOWER SEARCH LIMIT.
C     XR        --  THE SPECIFIED UPPER SEARCH LIMIT.
C     F         --  THE SPECIFIED REDUCTION IN THE INTERVAL
C                   OF UNCERTAINTY.
C     XBIG      --  THE X VALUE CORRESPONDING TO THE MAXIMUM Y VALUE.
C     YBIG      --  THE MAXIMUM Y VALUE FOUND DURING THE SEARCH.
C     XLF       --  THE FINAL LOWER LIMIT OF UNCERTAINTY.
C     XRF       --  THE FINAL UPPER LIMIT OF UNCERTAINTY.
C     K         --  MONITOR PRINT CONTROL.  K=0, NO MONITOR WILL
C                   BE PRINTED.  K=1, A SEARCH MONITOR WILL BE PRINTED.
C     N         --  THE NUMBER OF Y EVALUATIONS PERFORMED.
C                   N=0 IF AN ERROR IS DETECTED IN GOLD1.
C
C     THE USER MUST SUPPLY SUBROUTINE MERIT1 (X,Y) WHICH EVALUATES
C     THE FUNCTION BEING SEARCHED AT X AND RETURNS THE RESULT IN Y.
C
      N=0
      IF (XR.GT.XL) GO TO 107
      WRITE(NW,106)XL,XR
  106 FORMAT(41H *****ERROR MESSAGE SUBROUTINE GOLD1*****,/,5H XR =,
     1E15.7,21H NOT SMALLER THAN XL=,E15.7)
      GO TO 40
  107 IF (F.GT.0.0.AND.F.LT.1.0) GO TO 111
      WRITE(NW,110)F
  110 FORMAT(41H *****ERROR MESSAGE SUBROUTINE GOLD1*****,/,5H  F =,
     1E15.7,31H DOES NOT LIE BETWEEN 0. AND 1.)
      GO TO 40
  111 IF (K.NE.0) WRITE (6,33)
   33 FORMAT(37H1CONVERGENCE MONITOR SUBROUTINE GOLD1,//,58H      N
     1 Y1               Y2                X1                X2,//)
```

```
      XLF=XL
      XRF=XR
      DELTA=XRF-XLF
      SPAN=F*DELTA
      X1=XLF+0.381966*DELTA
      X2=XLF+0.618034*DELTA
      CALL MERIT1 (X1,Y1)
      CALL MERIT1 (X2,Y2)
      N=2
    3 IF (K.NE.0) WRITE (6,35) N,Y1,Y2,X1,X2
   35 FORMAT(I5,4(1X,E15.7))
      DELTA=XRF-XLF
      IF (DELTA.LE.SPAN) GO TO 4
      DELTA=0.618034*DELTA
      IF (Y1.GT.Y2) GO TO 2
      XLF=X1
      X1=X2
      Y1=Y2
      X2=XLF+0.618034*DELTA
      CALL MERIT1 (X2,Y2)
      N=N+1
      GO TO 3
    2 XRF=X2
      Y2=Y1
      X2=X1
      X1=XLF+0.381966*DELTA
      CALL MERIT1 (X1,Y1)
      N=N+1
      GO TO 3
    4 IF (Y2.GT.Y1) GO TO 6
      YBIG=Y1
      XBIG=X1
      GO TO 7
    6 YBIG=Y2
      XBIG=X2
    7 IF (K.NE.0) WRITE (6,38) XL,XR,F,YBIG,XBIG,XLF,XRF,N
   38 FORMAT(//,
     154H LEFTHAND ABSCISSA OF INTERVAL OF UNCERTAINTY .........,E15.7,/,
     254H RIGHTHAND ABSCISSA OF INTERVAL OF UNCERTAINTY ........,E15.7,/,
     354H FRACTIONAL REDUCTION OF INTERVAL OF UNCERTAINTY ......,E15.7,/,
     454H MAXIMUM ORDINATE DISCOVERED DURING SEARCH ............,E15.7,/,
     554H ABSCISSA OF MAXIMUM ORDINATE .........................,E15.7,/,
     654H NEW LEFTHAND ABSCISSA OF INTERVAL OF UNCERTAINTY .....,E15.7,/,
     754H NEW RIGHTHAND ABSCISSA OF INTERVAL OF UNCERTAINTY ....,E15.7,/,
     854H NUMBER OF FUNCTION EVALUATIONS EXPENDED IN SEARCH ....,I15,//)
   40 RETURN
      END
C
```

```
C
      SUBROUTINE WEIPLT(A,B,C)
      INTEGER CHAR(2)
      DIMENSION X(102),Y(102),ABSCA(14),XLAB(14),KAXIS(101),ORDIN(11)
      DIMENSION YLAB(11)
      DIMENSION FAIL(100),RANK(100)
      COMMON /SENSE/FAIL,RANK,NFAIL
      DATA CHAR(1),CHAR(2)/1H*,1H./
      DATA IDASH,IPLUS,IBLANK,II/1H-,1H+,1H ,1HI/
      DATA XLAB/0.1,1.0,10.0,20.0,40.0,60.0,80.0,
     *90.0,95.0,98.0,99.0,99.5,99.8,99.9/
C
      SX(D1)=-ALOG( ALOG( 1./(1.-D1)))
      SY(D2,D3)=ALOG(D2-D3)
      FX(D4,D5,D6,D7)=D6*ALOG( (D4-D5)/(D7-D5))
      YMIN=SY(C,A)-SX(0.01)/B
      YMAX=SY(C,A)-SX(0.999)/B
C
      DO 9 I=1,11
      Z=I-1
      ORDIN(I)=(YMAX-YMIN)*Z/10.0+YMIN
9     YLAB(I)=EXP(ORDIN(I))+A
C
      XMAX=SX(0.001)
      XMIN=SX(0.999)
C
      DO 5 I=1,NFAIL
      NFP1MI=NFAIL+1-I
      IF( RANK(I) .GE. 0.999) RANK(I)=0.999
      IF( RANK(I) .LE. 0.01) RANK(I)=0.01
      X(NFP1MI)=SX(RANK(I))
      Y(NFP1MI)=SY(FAIL(I),A)
5     CONTINUE
C
101   FORMAT(7X,F6.1,2X,101A1)
102   FORMAT(15X,101A1)
103   FORMAT(////,50X,'LN (LIFE-MINIMUM LIFE)',/)
104   FORMAT(/,50X,'RELIABILITY VERSUS LIFE')
105   FORMAT(9X,11(F10.2))
106   FORMAT(9X,11(F10.0))
C
      YSHFT=YMIN*100.0/(YMAX-YMIN)
      DO 6 I=1,14
      ABSCA(I)=SX( 1.0-(XLAB(I)/100.0) )
6     CONTINUE
      STEPX=(XMAX-XMIN)/100.
      KDELX=1
      LINE=1
      IND=0
      KLINE=1
C
10    IND=IND+1
C
```

```
      KSTEP=(X(IND)-XMIN)/STEPX+1.5
      TEMPY=Y(IND)*100.0/(YMAX-YMIN)-YSHFT
11    IF(KLINE .GT. LINE) GO TO 18
C
      DO 13 I=2,100
13    KAXIS(I)=IDASH
      DO 14 I=1,101,10
14    KAXIS(I)=IPLUS
C
      FXNOW=STEPX*(1.5-FLOAT(LINE))-XMIN
      F=A+(C-A)*EXP(FXNOW/B)
      F=SY(F,A)
      TEMPF=F*100.0/(YMAX-YMIN)-YSHFT
      KF=TEMPF+1.5
      KAXIS(KF)=CHAR(2)
      IF(KSTEP .GT. LINE) GO TO 16
      K=TEMPY+1.5
      KAXIS(K)=CHAR(1)
16    PRINT 101,XLAB(KDELX),(KAXIS(J),J=1,101)
C
      IF(KDELX .GE. 11) GO TO 27
C
      KDELX=KDELX+1
      KLINE=(ABSCA(KDELX)-XMIN)/STEPX+1.5
      GO TO 24
C
18    DO 19 I=2,100
19    KAXIS(I)=IBLANK
      DO 20 I=1,101,10
20    KAXIS(I)=II
      FXNOW=STEPX*(1.5-FLOAT(LINE))-XMIN
      F=A+(C-A)*EXP(FXNOW/B)
      F=SY(F,A)
      TEMPF=F*100.0/(YMAX-YMIN)-YSHFT
      KF=TEMPF+1.5
      KAXIS(KF)=CHAR(2)
      IF(KSTEP .GT. LINE) GO TO 33
      K=TEMPY+1.5
      KAXIS(K)=CHAR(1)
33    PRINT 102,(KAXIS(J),J=1,101)
C
24    LINE=LINE+1
      IF(LINE.GT. 102) GO TO 27
25    IF(KSTEP .GE. LINE)  GO TO 11
      IF(IND.LT.NFAIL) GO TO 10
      KSTEP=105
      GO TO 11
27    CONTINUE
C
```

```
      PRINT 106,(YLAB(J),J=1,11)
      WRITE(6,104)
C
      CALL OUTPUT(A,B,C)
C
      RETURN
      END
C
C
      SUBROUTINE SORTEM(A,N)
      DIMENSION A(N)
      NM1=N-1
      DO 8 I=1,NM1
      IP1=I+1
      DO 8 K=IP1,N
      IF( ABS(A(K)) .GE. ABS(A(I)) ) GO TO 8
      ATEMP=A(I)
      A(I)=A(K)
      A(K)=ATEMP
8     CONTINUE
      RETURN
      END
```

MERADCOM/OSU HYDRAULIC SYSTEM RELIABILITY PROGRAM

SECTION II

HYDRAULIC CYLINDERS

PREPARED BY PERSONNEL OF

FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA

May 1979

FINAL REPORT

June 1978 – May 1979

PREPARED FOR

U.S. ARMY MOBILITY EQUIPMENT RESEARCH
AND DEVELOPMENT COMMAND

Fort Belvoir, Virginia 22060

U.S. ARMY MERADCOM

HYDRAULIC SYSTEM RELIABILITY PROGRAM

HYDRAULIC CYLINDERS

FINAL REPORT

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PREFACE

This report presents a detailed account of the project activities concerning hydraulic cylinder specifications. Included are an OSU proposed revision of Army cylinder specification MIL-L-52762. Also included is a summary of an industrial survey conducted on hydraulic cylinders and discussion and justification of the proposed specification and test procedures.

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CHAPTER I

INTRODUCTION

The ultimate goal of the MERADCOM-OSU Hydraulic System Reliability Program is to exploit technology which has been developed and to apply this technology to induce more reliable mobile hydraulic systems on government-procured machinery. During previous MERADCOM-OSU programs, many test concepts and procedures for measuring the performance of hydraulic components were introduced to industry. The primary emphasis during this phase of the Hydraulic System Reliability Program was to obtain further industrial acceptance of those concepts and procedures in performance areas where no recognized standards exist. The availability of a full range of industrially-approved test procedures for hydraulic components would end the need for the military to require "military-special" performance tests. Many manufacturers of quality cylinders are reluctant to perform such special tests for one buyer, but could justify the expense of a full range of performance tests if such tests were universally acceptable and requested by customers other than military.

This phase of the MERADCOM-OSU project was concerned with test methods and specifications for hydraulic cylinders. In particular, project personnel were to review the current specification for double

acting hydraulic cylinders, MIL-C-52762 dated 19 September 1973 and attempt to update the performance test methods and requirements with current state-of-the-art technology.

The initial effort was to review the existing cylinder specification and critique each of the performance evaluation methods. Existing national and international standards were also studied. An industrial survey was conducted of both users and manufacturers of mobile hydraulic cylinders. The results of these efforts were compiled into a proposed set of cylinder performance evaluation procedures. These ideas were presented to recognized national standard organizations for suggestions, further development, and approval.

The majority of the cylinder performance test procedures are relatively straight-forward and have been utilized extensively in industry. Primary emphasis during this project was placed upon the standardization of all tests, and on the refinement of a valid and repeatable piston seal leakage test and cyclic endurance test procedure.

The following chapters summarize the project effort including the survey results and experimental test data. Finally, a set of test procedures and recommended acceptable parameter values is proposed for inclusion in a hydraulic cylinder specification.

CHAPTER II

CRITIQUE OF CURRENT CYLINDER SPECIFICATION

The current MERADCOM specification for double-acting hydraulic cylinders, MIL-C-52762 (ME) dated 19 September 1973, contains a total of eight performance evaluation procedures. These include the following:

1. Proof Pressure
2. Piston Drift
3. Piston and Rod Packing Drag (Packing Drag)
4. Cyclic Endurance (Stroke Cycle)
5. Impulse Endurance (Pressure Cycle)
6. Salt Spray Test
7. Abrasive Test
8. Low Temperature Test

A brief discussion of each of the test methods, their advantages and limitations, and related previous investigations is given below:

<u>Proof Pressure</u>

The proof pressure test consists of positioning and mechanically holding the piston at the approximate midpoint of the cylinder. Both

sides of the piston are then filled with oil. With the cap end port opened, apply an oil pressure equal to twice the working pressure to the rod end of the piston for a period of 60 seconds. With the rod end port opened, apply an oil pressure equal to twice the working pressure to the cap end of the piston for a period of 60 seconds. Any evidence of external leakage or deformation shall constitute failure of this test.

The proof pressure test is basically a fabrication or structural integrity test. It is primarily designed to evaluate the ability of the cylinder to withstand excessive pressure above the rated pressure with a certain safety factor in mind. This test is quite simple to perform and is generally accepted by most of industry in the form that it exists within the specification.

Piston Drift

The piston drift test consists of positioning and holding the piston of the cylinder at the approximate midpoint of the cylinder. Both sides are then filled with oil. The cap end port is capped and the rod end port is vented to atmosphere. The cap end of the slave cylinder is pressurized to attain a minimum of specified working pressure at the head end of the test cylinder. This pressure is maintained at an oil temperature of 65°C (150°F), plus ±2°C, (5°F) for 15 minutes. The length of travel of the piston during this period is measured and recorded. The above test is then repeated by capping

the rod end port and venting the cap end port to atmosphere then pressurizing the rod end of the slave cylinder. Piston drift greater than 0.64 centimetres (.25") per 15 minutes shall constitute failure of this test.

One of the primary functions of the hydraulic cylinder is to hold the position of a load for some finite period of time. Many times a cylinder is used to position a load such that other work can be accomplished, and there is only a small distance which the load can move during that period. The piston drift test, sometimes called position hold, was developed to evaluate the ability of a cylinder to accomplish this function. The piston drift test was examined during previous MERADCOM-OSU programs [ref. 1, 2] and it was concluded that unless special precautions were taken, repeatability of the test results is difficult to obtain. In fact, in reference [2], it was suggested that the piston drift or position hold test be replaced by a dynamic piston seal leakage test. The dynamic leakage test is somewhat controversial; however, because it is difficult to run and requires that one side of the cylinder be run dry which may subject the cylinder to abnormal lubrication and heating characteristics.

Several tests were run during this study on static piston seal leakage type tests including both the piston drift or position hold test and pressure decay. Experimental test results and a discussion of these methods is presented in a later chapter.

Piston and Rod Packing Drag (Packing Drag)

The packing drag test is a measure of the mechanical efficiency of the hydraulic cylinder. The pressure required to initiate rod motion, sometimes called 'break-away pressure', is an indication of the static friction characteristics of a cylinder. This is one of the primary parameters measured in this procedure. In order to conduct the packing drag test, the piston is positioned at the approximate mid-point of the test cylinder. Both sides of the cylinder are filled with oil and the cap end is vented to atmosphere. The pressure on the rod end of the cylinder is gradually increased until the piston begins to move. The pressure required to initiate movement is recorded as the break-away pressure. The pressure required to keep the cylinder in motion is also recorded. The above test is then repeated by pressurizing the cap end of the cylinder and venting the rod end. The cap end pressure and the rod end pressure exceeding values specified in Table 2-1 to actuate the piston constitute failure of this test.

Table 2-1. Breakaway Pressure

Bores, mm (inches) Stroke to 1.8 m (6 feet)	Breakaway Pressure, bar (psi) Rod End	Cap End
25-49 (1.0 - 1.9)	2.76 (40)	2.07 (30)
50-100 (2.0 - 3.9)	2.41 (35)	1.72 (25)
101-200 (4.0 - 7.9)	2.07 (30)	1.38 (20)
201-360 (8.0 - 14.0)	1.72 (25)	1.03 (15)

The packing drag test is relatively non-controversial and is generally accepted by most of industry. The break-away pressure or static friction

characteristics of a cylinder are directly influenced by the piston and rod seals. The preload on the seals designed into the cylinders is the determining factor for this parameter. Because of this fact, it is felt that the packing drag test is an important parameter which should be measured on a cylinder seal assembly.

Cyclic Endurance (Stroke Cycle)

The stroke cycle test consists of cycling the test cylinder under full stroke at a minimum of 20 cycles per minute at maximum operating pressure. The total numbers of cycles for this endurance test is based upon the classification of the cylinder which will be discussed later. All temperatures are maintained at 105°C (220°F) ±2°C (5°F) during the first 10% of the stroke cycles and at 65°C (150°F) ±2°C (5°F) during the remainder of the test run. The pressure, measured at the cylinder ports, shall rise from minimum to specified working pressure in a time interval not to exceed 0.05 seconds. The total external rod leakage accumulated during the entire test cycle is measured and recorded. Upon completion of the stroke cycle test piston drift and packing drag tests are redone. Malfunction prior to completion of the specified cycles, failure to meet the criteria set forth in the piston drift and packing drag test, or evidence of external leakage or damage shall constitute failure of this test. Rod packing leakage in excess of 2 millilitres per 1,000 cycles shall also constitute a failure.

The stroke cycle test can be a rather lengthy test because of the requirement for the cylinder to go full stroke for a large number of

cycles. The requirement of a minimum 20 cycles per minute can also be quite excessive if the cylinder has a long stroke length. As an example, consider a cylinder with a one-metre stroke or a two-metre cycle length. At a cycle speed of 20 cycles per minute the rod velocity would be 40 metres per minute. Typically, maximum rod velocities utilized in most testing are in the range of 15 metres per minute or 50 feet per minute. The excessive flow rate that is required to produce a 40 metres per minute cycle rate for the example cylinder would create high localized temperatures at the seals and also excessive pressure drops at the inlets and outlets of the cylinder.

A test somewhat similar to the stroke cycle test in the current MERADCOM specification was proposed in references [1] and [2]. This proposed test consisted in stroking the cylinder at 15% of the full stroke length at a rate of 15 metres per minute. The Society of Automotive Engineers Committee dealing with hydraulic cylinder standardization is currently considering an adoption of a version of the latter stroke cycle test.

Impulse Endurance (Pressure Cycle)

In order to conduct a pressure cycle test the piston rod is positioned at the approximate mid-point of the stroke. Pressure is alternately cycled between the cap end and the rod end from minimum pressure to a pressure limited to 125% of the working pressure at a minimum rate of 30 cycles per minute. The pressure measured at the cylinder ports shall rise from a minimum to a 125% working pressure at a time

interval not to exceed 0.05 seconds. The total number of cycles to be collected on the cylinder is based upon the classification of the cylinder which we will discuss later in this chapter. Upon completion of the required cycling, the piston drift test and packing drag test are repeated. Malfunction prior to completion of the specified cycles, failure to meet the criteria set forth in the drift and drag tests or evidence of external leakage or damage shall constitute failure of this test.

Because of the speed at which the pressure cycle test can be conducted, the total test time required for conducting this type of endurance test can be significantly less than for a stroke cycle test for long stroke lengths. A primary disadvantage of the pressure cycle test is the fact that the cylinders seals themselves are not being exercised in a manner similar to field service. For structural fatigue characteristics, however, the pressure cycle test can give valid data.

Salt Spray Test

The salt spray or corrosion-resistance test was developed to determine the ability of a hydraulic cylinder to operate in extremely harsh atmospheric conditions. Since the rod of the hydraulic cylinder is exposed to both the fluid environment inside the cylinder and the ambient environment, it should be capable of resisting deterioration due to this exposure. The salt spray test is conducted by subjecting the system rod to a 5% sodium chloride solution salt spray test in

accordance with ASTMB-117 for a period of not less than 30 hours. The piston rod surfaces are examined for corrosion at the conclusion of the test. Staining or corrosion that cannot be removed by rubbing with an oily rag or evidence of pitting shall constitute failure of this test.

It is felt that the salt spray test is an important test and should be included in a complete cylinder specification. However, the failure criteria specifying the current MERADCOM specification dealing with staining or corrosion that cannot be removed by rubbing with an oily rag is rather subjective and it is felt that a more quantatative measurement should be utilized to evaluate the failure criteria for this test. Definite noticeable pitting or excessive rod seal external leakage could be utilized as a valid failure criteria.

Abrasive Test

The abrasive test method in the current MERADCOM specification consists of subjecting the piston rod to a dusty environment in accordance with MIL-STD-810, test method 510 (controlled atmosphere dust box). The piston rod is cycled for 10 hours at a rate of 20 cycles per minute. At the conclusion of the test, the piston rod is examined for wear and damage. A packing drag test is then conducted. Upon completion of the packing drag test the cylinder is cycled for full stroke for 1,000 cycles. An external rod leakage is measured and recorded. Failure to meet the minimum criteria specified for packing drag, external rod leakage and visible wear or damage shall constitute failure of the test.

It is felt that the abrasive test, as it exists in the current specification, requires much more strictly-controlled conditions than are necessary. It was the opinion of the investigators that an abrasive test, as presented here, was very rarely used in industry. In fact, the industrial survey presented in the next chapter verified this conclusion. Since the abrasive test is primarily the measurement of the ability of rod and wiper seals to withstand contamination, it is felt that the utilization of the standard wiper seal test procedure which has been developed during previous MERADCOM-OSU programs could be utilized in place of the abrasive test. The wiper seal test also utilizes a dust box, or dusty atmosphere, but the measurements or performance criteria specified were more readily determinable. A complete discussion of the wiper seal test is presented in a later chapter.

Low Temperature Test

The low temperature test method consists of operating the cylinder at -32°C (-25°F) after storage and at -35°C (-30°F) without the piston binding or a change in speed of cylinder stroking. When specified, the cylinder shall operate satisfactorily at -46°C (-50°F) after storage at the same temperature without piston bonding or change in the speed of cylinder stroking.

The low temperature test is probably the one test that is a "military special" requirement. If it is necessary to have cylinders

operated at the extreme cold temperatures indicated in this specification, then it is felt by the investigators that such a low temperature test is probably a valid requirement. The failure criteria of change in cylinder stroking speed is a rather arbitrary determination and it is felt that significant improvements could be made.

General Test Requirements

A rating system is included in the current military specification dealing with the expected service life of the hydraulic cylinder. Three types of classes are established as follows:

Type 1. 700,000 duty cycles for heavy-duty service

Type 2. 200,000 duty cycles for moderate duty-service

Type 3. 100,000 duty cycles for light duty-service

It is the opinion of the investigators that such a rating system is required for hydraulic cylinders because of the specialized service to which individual cylinders are subjected. The foundation for the establishment of the exact duty cycle numbers given above is not known to the investigators. However, it is felt that many industrial cylinder manufacturers and users use somewhat similar classification systems.

The hydraulic fluid utilized for most of the tests in the current military specification is MIL-L-2104, grade 10. This oil is readily available and widely used in the fluid power industry today and thus, it is recommended that the oil remain as the primary test oil for the specification. A test temperature of 65°C (150°F) is currently

utilized for most of the test procedures. It is recommended that this temperature be changed to reflect the current trends of the SAE's hydraulic cylinders standardization committee.

CHAPTER III

SURVEY RESULTS AND DISCUSSIONS

Before any changes in Army cylinder specifications were suggested, it was necessary that a survey be conducted. The results of a survey of industrial opinion with regard to the Army's hydraulic cylinder requirements would allow the optimum course to be taken in modifying the various specifications, including the specified tests and test procedures to more closely conform to current industrial practice. The FPRC personnel are cognizant of many of the present cylinder testing and evaluation standards, having fostered many of them. Industrial input on various aspects of the Army's own specifications, when combined with knowledge of current cylinder testing practice and standards, made an excellent base from which hydraulic cylinder specifications could be reviewed and revised.

The questionnaire was conducted by the U.S. Army MERADCOM/OSU FPRC team during their effort to update hydraulic cylinder test methods. The "basics" of the tests were evaluated as well as specifics, such as pass-fail criteria, test temperatures, and speeds. A summary of the questionnaire was sent to respondents, see Appendix A. Several lengthy comments were included with the returned surveys and will be discussed as appropriate. A total of 131 questionnaires were sent, thirty-eight (29%) were returned. Also, a number of letters were received indicating

that the questionnaire would not be returned for various reasons. These are not included as respondents. Hydraulic cylinder manufacturers responding totaled twenty-one, seven questionnaires were received from companies who primarily bought and/or used hydraulic cylinders, ten of those responding were in the "Both" group, so called because they both used and manufactured hydraulic cylinders, primarily for the use on equipment they sell. The questions posed and a summary of the answers received, including various comments, are presented below.

1. <u>The following "rating" system is used in current military specifications:</u>

   "Expected Service Life"

   Type I - 700,000 Duty Cycles (Heavy Duty)

   Type II - 200,000 Duty Cycles (Moderate Duty)

   Type III - 100,000 Duty Cycles (Light Duty)

1a. <u>Is it necessary to classify cylinders in this way? "Heavy duty," "Moderate duty," etc. by expected service life?</u>

	□Yes	□No	Comments:
Users		7 Yes	0 No
"Both"		6 Yes	4 No
Manufacturers		8 Yes	13 No

Many of the "No" answers included comments indicating a need for a more precise definition of the "duty cycle" to be used in life determinations. Some companies commented that the Type I life was too short

and that their cylinders were designed to operate well in excess of 700,000 cycles. Additional classes including one million and ten million duty cycles were also suggested.

1b. <u>Does your company/division classify hydraulic cylinders in this way?</u>

□ Yes □ No

Users	1 Yes	6 No
"Both"	5 Yes	5 No
Manufacturers	2 Yes	19 No

The response to this question was not what might be expected in light of the response to question 1a. Comments indicated that hydraulic cylinders were generally tailored to the specific application rather than designed for some number of "duty cycles."

1c. <u>If yes, does your company/division use the above ratings exactly?</u>

□ Yes □ No

Users	0 Yes	3 No
"Both"	1 Yes	5 No
Manufacturers	0 Yes	10 No

The only yes response included a comment that this was "for design purposes only." Three of the "Both" group indicated that they

used a similar rating system. The manufacturers generally used the Light duty, Medium or Moderate duty, and Heavy duty terms to distinguish between design operating pressure ranges. Designing for the application was occasionally stressed.

The proof pressure test was described, including pass/fail criteria, and the following questions were presented.

2a. <u>My company/division uses the above test, or similar.</u>

□Frequently □Seldom □Never

Users	3 Frequently	2 Seldom	2 Never
"Both"	5 Frequently	3 Seldom	2 Never
Manufacturers	10 Frequently	7 Seldom	4 Never

The word "similar" in the question was occasionally underlined by those surveyed. Some commented that they proof tested the cylinder at the ends of its stroke.

2b. <u>Is the proof pressure always twice the working pressure?</u>

□Yes □No, it is _____ times the working pressure.

Users	2 Yes	5 No
"Both"	2 Yes	7 No
Manufacturers	1 Yes	19 No

Comments indicated that there might be a difference between the proof pressure used on production cylinders and that used to qualify

a design. Pressure multipliers as high as 4 were suggested; the majority were either 1.6 or 1.5, the lowest given was 1.25. Some suggested different factors for different types of service, especially shock vs. non-shockloading.

2c. <u>A proof pressure test is needed in any procurement specification.</u>

□Agree □Disagree

Users	6 Agree	1 Disagree
"Both"	7 Agree	3 Disagree
Manufacturers	17 Agree	4 Disagree

Some of those disagreeing with this statement indicated that this was due to its all inclusive nature. From the response of the manufacturers, it is apparent that many routinely do some form of pressure test on their production cylinders.

The piston drift measurement procedure was presented; the statements below were posed and the following responses were received.

3a. <u>My company/division uses the above test, or similar.</u>

□Frequently □Seldom □Never

Users	4 Frequently	2 Seldom	1 Never
"Both"	6 Frequently	2 Seldom	1 Never
Manufacturers	8 Frequently	8 Seldom	3 Never

3b. <u>The test temperature of 65°C (150°F) is reasonable and proper.</u>

□ Yes   □ No, a temperature of _______ would be superior for such tests.

In many of the comments, attention was drawn to the difficulties in maintaining 65 ±2°C over a period of 15 minutes. The importance of thermal expansion/contraction and its effect on apparent drift measurements were also discussed. A number of those responding suggested that a lower temperature would be easier to maintain and provide safer and easier handling of test specimens and equipment.

3c. <u>The test time of 15 minutes is reasonable and proper.</u>

□ Yes   □ No, a test time of _____ minutes would be better.

Users	4 Yes	3 No
"Both"	0 Yes	10 No
Manufacturers	8 Yes	12 No

Many of those responding felt that a shorter test time, generally in the neighborhood of 5-10 minutes, would be better for production tests. Some suggested as little as one minute, while another considered 30 minutes appropriate. One response indicated that they preferred to measure piston drift after the cylinder was installed on the machine.

3d. <u>It is necessary to measure piston drift in both directions.</u>

□ Agree   □ Disagree

Users	6 Agree	1 Disagree
"Both"	9 Agree	1 Disagree
Manufacturers	14 Agree	7 Disagree

The question sparked discussion on the various types of seals available and the leakage characteristics of different designs. Other comments included the idea that drift can only occur in one direction in some applications and that, in this case, a bidirectional test was unnecessary.

3e. <u>Piston drift of 6.35mm (0.25 inches) per 15 minutes, 0.42mm (.0167 inches) per minute as the maximum acceptable, is a reasonable requirement for the intended use.</u>

□Agree □Disagree, I prefer:

Users	3 Agree	4 Disagree
"Both"	0 Agree	10 Disagree
Manufacturers	9 Agree	11 Disagree

Those who disagreed had widely varying opinions of how much drift should be allowed. One of the "Both" questionnaires indicated that several times the above rate was acceptable on certain earthmoving equipment applications. Some manufacturers considered the 6.35mm drift per 15 minutes requirement too strict, but other manufacturers suggested 0.5mm/15 min., 0.010"/hr., and "none" as the maximum drift acceptable. Discussion indicated that the allowable drift was strongly dependent on the particular application.

3f. <u>A test for piston drift under load is necessary for any cylinder procurement specification.</u>

□ Agree □ Disagree

Users	6 Agree	1 Disagree
"Both"	7 Agree	2 Disagree
Manufacturers	8 Agree	12 Disagree

The only user who disagreed commented that piston drift could be part of an assembled vehicle test. Some of those responding commented that piston drift testing, except in critical applications, did not need to be checked except on a sampling or perhaps a qualification basis. Leakage flow rate at end of stroke under pressure was suggested by a number of groups as a possible "production line" alternative.

Industrial opinions on the piston and rod packing drag test were then solicited. A table of "breakaway pressures" for various bores was included.

4a. <u>My company/division uses the above test, or similar.</u>

□ Frequently □ Seldom □ Never

Users	1 Frequently	5 Seldom	1 Never
"Both"	6 Frequently	4 Seldom	0 Never
Manufacturers	9 Frequently	5 Seldom	6 Never

4b. <u>My company/division specifies breakaway pressures that are generally _______ the above.</u>

□ Higher than □ About the same as □ Lower than

Users	2 Higher	3 About the same	0 Lower
"Both"	6 Higher	2 About the same	2 Lower
Manufacturers	4 Higher	5 About the same	5 Lower

A number of those responding neglected to mark this question; some indicated that they did not use such a test. Other comments included discussion on specifying a specific cylinder for a particular application, and the effects of pressure and time at rest on packing drag tests.

4c. <u>Breakaway pressure specifications, such as listed in Table 1 apply to:</u>

- ☐ Oversize and standard rods alike
- ☐ Standard rods only
- ☐ Any type of packing used
- ☐ Standard packings only

Users	1 Oversize	2 Standard rods
	2 Any packing	3 Standard packing
"Both"	4 Oversize	2 Standard rods
	5 Any packing	3 Standard packing
Manufacturers	5 Oversize	6 Standard rods
	5 Any packing	6 Standard packing

4d. <u>Breakaway pressures must be tested in both directions. (Assume rod and bore sizes are known.)</u>

☐ Agree ☐ Disagree

Users	5 Agree	2 Disagree
"Both"	10 Agree	0 Disagree
Manufacturers	18 Agree	2 Disagree

4e. <u>Breakaway pressures or some measure of packing drag is needed in any cylinder procurement specification.</u>

□Agree □Disagree

Users	4 Agree	3 Disagree
"Both"	10 Agree	0 Disagree
Manufacturers	6 Agree	15 Disagree

Some considered this test as pertinent concerning only special application cylinders. A few commented that this test would indicate poor fits and improper assembly; Cylinder orientation and the effects of rod weight were also mentioned.

Cyclic endurance testing procedure and pass/fail criteria were then presented for evaluation.

5a. <u>My company/division uses the above test, or similar.</u>

□Frequently □Seldom □Never

Users	5 Frequently	1 Seldom	1 Never
"Both"	5 Frequently	3 Seldom	2 Never
Manufacturers	8 Frequently	5 Seldom	8 Never

5b. The rate of 20 cycles per minute is reasonable and proper for most applications.

□Agree □Disagree

Users	2 Agree	5 Disagree
"Both"	3 Agree	7 Disagree
Manufacturers	7 Agree	13 Disagree

One respondent commented that cycle rate should be adjusted in relation to stroke length using some specified rod velocity. Another commented that their cylinders operated at less than 10 c.p.m. A significant number of those responding commented on the difficulties involved in cycling a large bore, long stroke cylinder at 20 c.p.m.

5c. Would the specification of maximum and/or minimum cylinder stroking speeds for endurance testing be preferable to the specification of a cycle rate?

□Yes □No

Users	6 Yes	1 No
"Both"	8 Yes	2 No
Manufacturers	15 Yes	5 No

Comments indicated that stroking speeds (rod velocity) were preferable, in general, provided that there was not excessive heat buildup in the cylinder and that the ports provided adequate flow.

5d. <u>Fifteen meters per second (50 ft/sec) has been suggested as a maximum stroking speed; is this reasonable and proper for endurance testing?</u>

□ Yes □ No

This question was stated incorrectly; it should have been meters per minute rather than per second. Response to the question as printed on the questionnaire is given below.

Users	3 Yes	4 No
"Both"	2 Yes	8 No
Manufacturers	8 Yes	9 No

Comments indicated that some of those responding "yes" had seen the error and were answering the question as it should have read. Others agreed saying that there was little, if any reason, to cycle faster than 15 meters per second.

5e. <u>It is necessary to cycle full stroke in cyclic endurance testing.</u>

□ Agree □ Disagree

Users	3 Agree	4 Disagree
"Both"	5 Agree	5 Disagree
Manufacturers	11 Agree	10 Disagree

Some of the comments in favor of full stroking were: cylinder should bottom out on at least one end, preferably the rod end, bore dimensions

may vary causing hot spots, full stroke is considered necessary for fatigue tests. Others commented: depends on application and end features, impulses are important for seals, full stroke too time consuming, and cylinders are seldom stroked fully.

5f. It is my opinion/experience that seals under cycling fail because of:

□Reversals in motion □Distance swept □Both □Neither

Users	0 Reversals	1 Distance	3 Both	2 Neither
"Both"	0 Reversals	0 Distance	7 Both	2 Neither
Manufacturers	3 Reversals	3 Distance	12 Both	3 Neither

Other causes for seal failure suggested were: contamination, temperature, pressure, dimensional variance, and cylinder internal surface finish incorrect.

5g. An elevated (105°C, 220°F) oil temperature for the first portion of a cyclic test is necessary and proper.

□Agree □Disagree

Users	3 Agree	4 Disagree
"Both"	3 Agree	6 Disagree
Manufacturers	3 Agree	18 Disagree

One user commented that the elevated temperature should be used throughout the test. A few commented that this temperature was above that for which their seals were rated. Some comments suggesting that the test be tailored to suit the specific application were also received.

5h. A temperature of 65°C (150°F) is proper for the majority of an endurance test.

	□ Agree	□ Disagree
Users	1 Agree	6 Disagree
"Both"	2 Agree	8 Disagree
Manufacturers	13 Agree	8 Disagree

A number of comments were received stating that 65°C is too high; a similar number, however, expressed a preference for a higher temperature. Application type testing was also mentioned.

5i. The pressure rise, from 0 to specified working, in 0.05 seconds is reasonable and proper for this type of test.

	□ Agree	□ Disagree
Users	6 Agree	1 Disagree
"Both"	3 Agree	6 Disagree
Manufacturers	10 Agree	11 Disagree

A significant number (6) of those disagreeing suggested a lower pressure rise rate. Many of the pressure rise rates suggested were in psi/sec rather than a fixed time period.

5j. My company/division measures rod leakage during cyclic endurance tests.

□Frequently □Seldom □Never

Users	4 Frequently	3 Seldom	0 Never
"Both"	8 Frequently	1 Seldom	1 Never
Manufacturers	18 Frequently	1 Seldom	1 Never

5k. My company/division measures, after cyclic endurance testing.

□Piston drift under load □Breakaway pressure on packing drag

□Other

Users	6 Piston drift	2 Breakaway	0 Other
"Both"	9 Piston Drift	3 Breakaway	0 Other
Manufacturers	12 Piston Drift	7 Breakaway	0 Other

Some of the other tests used were: low pressure (breakaway) test, external leakage, piston seal leakage, and rod leakage.

5l. A maximum acceptable rod leakage rate of 2.0mℓ per 1000 cycles is reasonable and proper, regardless of stroke length.

□Agree □Disagree

Users	2 Agree	5 Disagree
"Both"	3 Agree	6 Disagree
Manufacturers	8 Agree	10 Disagree

Six of those commenting on this question considered 2mℓ per 1000 cycles an excessive leakage rate; two manufacturers said that no measurable

leakage should be allowed. A few respondents considered stroke length to be a factor.

The impulse endurance (pressure cycle) test procedure was presented in detail along with pass/fail criteria.

6a. My company/division uses the above test, or similar.

□Frequently □Seldom □Never

Users	3 Frequently	2 Seldom	2 Never
"Both"	8 Frequently	0 Seldom	2 Never
Manufacturers	5 Frequently	9 Seldom	5 Never

6b. The following parameter values are reasonable and proper for an impulse endurance test.

Pressure 125% maximum of working

□Agree □Disagree

Users	5 Agree	1 Disagree
"Both"	3 Agree	7 Disagree
Manufacturers	15 Agree	3 Disagree

30 or more cycles per minute

□Agree □Disagree

Users	4 Agree	2 Disagree
"Both"	9 Agree	1 Disagree
Manufacturers	13 Agree	5 Disagree

Pressure rise in 0.05 seconds maximum

□ Agree □ Disagree

Users	5 Agree	1 Disagree
"Both"	5 Agree	4 Disagree
Manufacturers	12 Agree	6 Disagree

Two of those responding suggested a pressure higher than 125% of working, another two suggested 115% as preferable. Yet another two commented that the minimum cycle rate should be less than 30 c.p.m., allowing easier testing of large cylinders. Pressure rise comments were similar to those given with 5i above. As with many of the questions, the idea of tailoring testing to an application was discussed.

6c. A separate impulse endurance (pressure cycle) test of some form should be required even if stroke cycle testing is done.

□ Agree □ Disagree

Users	3 Agree	3 Disagree
"Both"	6 Agree	4 Disagree
Manufacturers	12 Agree	7 Disagree

6d. It is necessary to re-run the following tests after impulse endurance testing.

□ Piston drift □ Packing Drag □ Other

Users	4 Piston	1 Packing
"Both"	7 Piston	0 Packing
Manufacturers	9 Piston	4 Packing

Some of the other tests suggested were: internal leakage, rod leakage, structure leakage, performance tests, and proof pressure. Five did not consider it necessary to re-run any tests; they were interested primarily in metal fatigue and related failures.

The salt spray test was then presented.

7a. My company/division uses the above test, or similar.

□ Frequently □ Seldom □ Never

Users	2 Frequently	2 Seldom	3 Never
"Both"	3 Frequently	3 Seldom	4 Never
Manufacturers	3 Frequently	9 Seldom	9 Never

7b. The above test is necessary for most procurement specifications.

□ Agree □ Disagree

Users	4 Agree	2 Disagree
"Both"	8 Agree	2 Disagree
Manufacturers	4 Agree	15 Disagree

Comments indicated that if qualified rod material was used it would be unnecessary to test individual cylinders.

Only one comment was received on the possible ambiguity of the pass/fail criteria.

The procedure for the abrasive test was presented in detail including pass/fail criteria.

8a. <u>My company/division uses the above test, or similar.</u>

□ Frequently □ Seldom □ Never

Users	1 Frequently	2 Seldom	4 Never
"Both"	1 Frequently	2 Seldom	7 Never
Manufacturers	0 Frequently	5 Seldom	16 Never

8b. <u>This test is necessary and useful even if specific wiper seal data is available.</u>

□ Agree □ Disagree

Users	3 Agree	4 Disagree
"Both"	1 Agree	8 Disagree
Manufacturers	3 Agree	13 Disagree

Two commented that the usefulness of this test was dependent upon the application of the cylinder.

8c. <u>The dust environment specified is proper and reasonable for use with this test.</u>

□ Agree □ Disagree

Users	4 Agree	1 Disagree
"Both"	1 Agree	3 Disagree
Manufacturers	7 Agree	8 Disagree

Four of those who did not answer this question indicated that test method 810 (the dust environment specification used) was unfamiliar to them. Two commented that the proper contaminant environment for testing is dependent upon the proposed application of the cylinder. One respondent considered test method 810 impractical.

8d. <u>It is necessary that the following be evaluated after testing.</u>

□ Rod condition □ Packing drag □ Rod leakage

Users	6 Rod condition	3 Packing drag	7 Rod leakage
"Both"	2 Rod condition	1 Packing drag	5 Rod leakage
Manufacturers	11 Rod condition	4 Packing drag	12 Rod leakage

Other tests suggested were: internal contamination level, bore, piston and seal wear, and also head wear.

8e. <u>The above test, or similar, is necessary for use in procurement specifications.</u>

□ Agree □ Disagree

Users	3 Agree	5 Disagree
"Both"	3 Agree	5 Disagree
Manufacturers	3 Agree	14 Disagree

Five manufacturers commented that such a test was unnecessary for many applications. One user thought that this test could be done as part of the total vehicle test.

8f. Could a wiper seal test, such as SAE J1195 replace the abrasive test?

☐ Yes ☐ No

Users	4 Yes	1 No
"Both"	4 Yes	0 No
Manufacturers	12 Yes	1 No

One user suggested that a test using dirt or sand frozen to the rod be done with SAE J1195. Another brought out the point that even a "good" seal will operate poorly if improperly installed.

The low temperature test was presented, with pass/fail criteria.

9a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

Users	1 Frequently	3 Seldom	3 Never
"Both"	0 Frequently	3 Seldom	5 Never
Manufacturers	0 Frequently	4 Seldom	16 Never

9b. The above test, or similar, should be included in a procurement specification.

☐ Agree ☐ Disagree

Four commented that the test should be used only if the application dictates. One "both" commented that this particular test was "not very definitive of cold temperature functional requirements - seal performance, brittle fracture, etc. One manufacturer suggested that the test could be omitted if qualified seals were used.

9c. Is "change in stroking speed" a valid failure criterion?

□Yes □No, I prefer:

Users	1 Yes	4 No
"Both"	3 Yes	3 No
Manufacturers	1 Yes	14 No

Some respondents indicated that other parameters might be better for pass/fail judgments. Those suggested were: external leakage, especially around the rod, internal leakage and change in pressure required for stroking. One respondent indicated that most labs would be unable to run such a test.

There are a number of tests available for evaluating hydraulic cylinders other than those used by the military. Various aspects of some of these tests were evaluated in the latter part of the questionnaire.

The method of determining static internal leakage using the pressure decay technique was presented in summary.

10a. My company/division uses the above test, or similar.

□Frequently □Seldom □Never

Users	1 Frequently	2 Seldom	4 Never
"Both"	2 Frequently	2 Seldom	7 Never
Manufacturers	1 Frequently	5 Seldom	14 Never

10b. The bulk modulus of the test oil will be constant or can be controlled and measured from test to test.

□ Agree □ Disagree

Users	1 Agree	6 Disagree
"Both"	5 Agree	4 Disagree
Manufacturers	4 Agree	7 Disagree

One user agreed with the condition that aeration be carefully controlled. Another respondent agreed but commented that "much expensive equipment" would be required. One respondent, from the "both" category, gave a lengthy discussion of the sensitivity of the test to changes in fluid temperature and general nonrepeatability.

The contaminant wear test was briefly described; a piston drift test was specified to measure degradation.

11a. My company/division uses the above test, or similar.

□ Frequently □ Seldom □ Never

Users	0 Frequently	2 Seldom	5 Never
"Both"	0 Frequently	1 Seldom	9 Never
Manufacturers	0 Frequently	2 Seldom	18 Never

One manufacturer commented that, while they did not presently use a test like this, they had plans to do so in the future.

11b. <u>My company/division is concerned with cylinder degradation under contaminated conditions.</u>

□Yes □No

Users	7 Yes	0 No
"Both"	8 Yes	2 No
Manufacturers	15 Yes	3 No

11c. <u>The following tests should be run following the contaminant test above.</u>

□Piston Drift □Packing Drag □Dynamic Leakage (internal)

Users	2 Piston	2 Packing	5 Dynamic
"Both"	6 Piston	0 Packing	1 Dynamic
Manufacturers	10 Piston	4 Packing	7 Dynamic

Some of the other tests suggested were: external leakage, rod leakage and rod leakage per 1000 cycles.

11d. <u>The 300 mg/ℓ, ACFTD, is a reasonable and proper contamination level.</u>

□Agree □Disagree

Users	5 Agree	1 Disagree
"Both"	3 Agree	5 Disagree
Manufacturers	8 Agree	3 Disagree

One "both" commented that 100mg/ℓ might be superior, another of the "both" group indicated, in contrast, that in their experience 30,000

cycles at 300mg/ℓ produced no significant wear. Two commented that 300mg/ℓ was too high to be realistic.

11e. <u>The current plans utilize full distribution ACFTD only. Is this acceptable or are classified dust tests necessary?</u>

□Acceptable □Need cut-dust, why?

Users	6 Acceptable	1 Need cut-dust
"Both"	8 Acceptable	1 Need cut-dust
Manufacturers	8 Acceptable	1 Need cut-dust

The user who felt cut-dust tests were required indicated a desire for several options in the test. The two others who considered cut-dust necessary were anxious to tailor the test to a particular application. One of those suggesting full distribution ACFTD felt that this would yield superior wear data and best fit real world conditions.

11f. <u>Is this test necessary if a wiper seal (dust box) test is also conducted?</u>

□Yes □No

Users	3 Yes	3 No
"Both"	4 Yes	4 No
Manufacturers	7 Yes	9 No

Two of those answering yes commented that contaminated fluid is not always caused by wiper seal failure. One user commented that the use of proper filtration would help protect the cylinder in addition to pumps, valves, etc.

A proposed hydraulic cylinder cushion test using various applied forces was presented.

12a. My company/division uses the above test, or similar.

□Frequently □Seldom □Never

Users	0 Frequently	2 Seldom	4 Never
"Both"	0 Frequently	4 Seldom	4 Never
Manufacturers	4 Frequently	6 Seldom	10 Never

12b. My company/division uses/manufactures the following proportions of cylinders.

Fixed cushions	______% ≃ 5%	Rough average of those responding, may not be indicative of overall trends.
Adjustable cushions	______% ≃ 5%	
No cushions	______% ≃ 90%	

12c. Is a standard hydraulic cylinder cushion test method needed?

□Yes □No

Users	3 Yes	3 No
"Both"	5 Yes	5 No
Manufacturers	11 Yes	6 No

Comments indicated a desire for a broad based standard, allowing special evaluations as necessary.

12d. Is measurement of velocity versus displacement a valid technique for evaluation cylinder cushions?

□Yes □No

Users	4 Yes	1 No
"Both"	6 Yes	3 No
Manufacturers	10 Yes	4 No

Various techniques were suggested; they were: the G's upon entering the cushion and at end impact; velocity, pressure and flow vs. displacement; various pressure readings; use a stated load mass with a stated initial velocity; pressure vs. time and cylinder position.

Dynamic leakage, by the direct measure technique was presented for discussion.

13a. <u>My company/division uses the above test, or similar</u>

□Frequently □Seldom □Never

Users	2 Frequently	2 Seldom	3 Never
"Both"	2 Frequently	1 Seldom	7 Never
Manufacturers	6 Frequently	3 Seldom	11 Never

13b. <u>Dynamic leakage rate is an important cylinder parameter.</u>

□Yes □No

Users	6 Yes	1 No
"Both"	1 Yes	7 No
Manufacturers	4 Yes	16 No

13c. <u>The life and operation of the piston seal would not be affected by one side being run "dry."</u>

□Agree □Disagree

Users	1 Agree	6 Disagree
"Both"	0 Agree	10 Disagree
Manufacturers	5 Agree	15 Disagree

Six comments indicated that the effects of dry operation would vary with various seal types and materials.

Additional Comments

One user added a number of additional tests to the list. These were: 1. a buckling test; 2. a sustained tension test with pressure cycling to check integrity of end fittings, glands and base welds; and 3. a tension limit test, with the cylinder fully pressurized while extended and additional tension load of twice the rated load was applied and held for 3 seconds.

There were also occasional, general comments concerning the difficulties in creating a universal or standard test package and the problems that occur when such a standard is used to specify requirements for a particular application. Some of the most frequently mentioned problems were excessive costs and delays associated with over specification. The significant difference between "mobile" and "industrial" cylinders and applications were often noted. It is the authors' opinion that the two should be treated separately in future work.

CHAPTER IV
EXPERIMENTAL TEST RESULTS

The primary cylinder test procedures which were initially felt to require verification testing were an internal piston seal leakage test method, the cyclic endurance method and the proposed contaminant sensitivity technique. Because of time and financial constraints on this project, the pressure decay piston seal leakage test method was the only procedure upon which extensive testing was conducted. An attempt was made to conduct a cyclic endurance test; however, long delivery times for purchased components and numerous equipment failures precluded the collection of any significant test results.

Piston Seal Internal Leakage

There are a number of existing test methods available for evaluating the leakage around a cylinder piston seal. These include actual static leakage measurement, dynamic leakage measurement and the piston drift or position hold test. In addition, a test called the pressure decay was recently developed for an experimental study with a major mobile equipment manufacturer.

Previous experience with the actual static leakage measurement has resulted in extremely poor discrimination and test repeatability.

This is generally due to the low quantities of leakage past most piston seals under static conditions. The piston drift test, discussed in Chapter I, is also subject to error because of the small piston or rod movement with most seal packages. The dynamic leakage test, so called because the piston is cycled during the test, is not conducted under realistic conditions. One side of the cylinder is vented to atmosphere during cycling and, therefore, is run "dry" resulting in abnormal lubrication. Furthermore, this test is not representative of a parameter of importance in an actual application; whereas, the piston drift is certainly critical. Finally, although the pressure decay test at first appears to be a valid indicator of static leakage, there are several shortcomings and parameters which must be precisely controlled in order to obtain repeatability. It was initially felt that the pressure decay test would produce representative test data and a separate test program was conducted to verify the repeatability and applicability of this test.

Pressure Decay Technique

The pressure decay test is conducted in accordance with the following procedure:

1. Cycle the test cylinders under rated pressure to displace entrained air and to establish the required test temperature.
2. Position the test cylinder as desired (usually midstroke).
3. Physically block the test cylinder, then pressurize the cap end and vent the rod end.

4. Instantaneously close a valve on the pressure supply and record the decay in the internal pressure in the cylinder end.
5. Repeat the above with the rod end pressurized and the cap end vented.
6. Repeat the above sequence at a pressure equal to ten percent of rated pressure.

The data collected during a pressure decay test are graphs of the decrease in cylinder pressure versus time. Typical pressure decay curves are shown in Fig. 4-1. These curves alone can give some indication of seal performance; however, for comparative purposes, it is necessary to estimate an equivalent leakage rate from these pressure decay values.

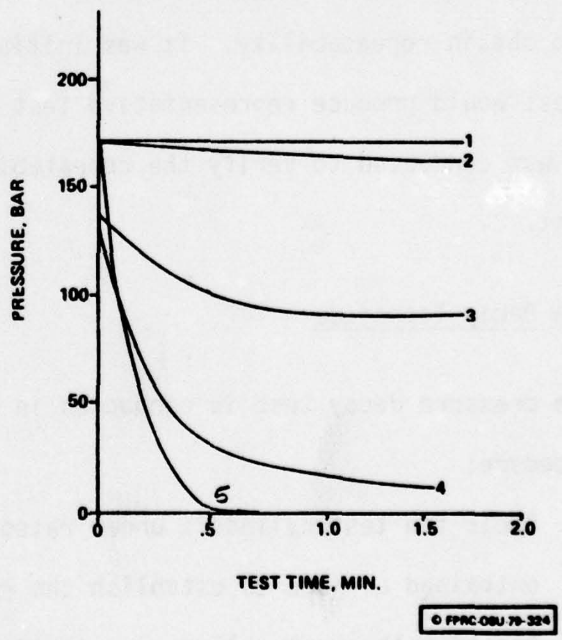

Fig. 4-1. Pressure Decay Characteristics of Typical Piston Seals.

For a compressible fluid, the equation describing compressibility can be written as follows:

$$\frac{dV}{dt} = \frac{Vo}{\beta} \frac{dP}{dt}$$

where: V = fluid volume

t = time

Vo = initial fluid volume

β = bulk modulus

P = fluid pressure

For a cylinder test, the initial fluid volume and compressibility can be estimated and values of dP/dt can be determined from the experimental pressure decay data. Thus, values of dV/dt or equivalent leakage can be calculated.

In order to determine the value of dP/dt for a given cylinder test, the experimental pressure decay data were fitted to an exponential curve using least squares techniques. The initial value for dP/dt was then estimated for each test by calculating the first derivative of the exponential pressure versus time equation at a value of t = 0. This corresponds to the rate of pressure change when the pressure is a maximum.

A total of twenty cylinders were evaluated with the pressure decay test. The resulting data were converted by using the above technique to equivalent leakage rate. A fluid bulk modulus, β, of 100,000 was

assumed for the conversion. The data from the tests are presented in the following table:

Table 4-1. SUMMARY OF PRESSURE DECAY DATA.

Cylinder No.	Equivalent Leakage Rate (mℓ/min)			
	17.2 BAR (250 PSI)		172 BAR (2500 PSI)	
	Rod End	Cap End	Rod End	Cap End
A1	80.0	94.2	908	1070
A2	0.67	0.20	2.11	1.75
A3	80.0	94.2	11.3	10.7
A4	4.67	94.2	2.21	6.26
B1	0.0	92.6	1.30	0.87
B2	--	4.82	--	2.51
B3	69.5	3.49	1.84	3.43
B4	2.52	92.6	1.69	2.45
C1	1.39	5.83	3.21	3.91
C2	2.82	20.8	2.73	5.25
C3	91.7	122	6.35	11.8
C4	2.50	122	2.39	7.21
D1	100.5	23.3	8.42	121
D2	3.05	121	2.54	123
D3	80.0	5.59	7.21	5.69
D4	5.92	94.2	2.11	95.9
E1	62.4	89.2	2.84	3.57
E2	3.73	89.2	1.65	5.46
E3	--	56.3	--	385
E4	9.24	107	1.99	15.1

The cylinder data in the above table are for five different cylinder models, (represented by the letters A-E), and four identical cylinders in each group, (represented by the numbers 1-4). Data were collected for 17.2 bar and 172 bar pressures with both the cap and rod ends pressurized.

It can easily be observed from the test data expressed as equivalent leakage rates that the pressure decay results are quite erratic. There appears to be no repeatability for identical cylinders.

One possible explanation for such poor repeatability is the assumption of a constant bulk modulus equal to 100,000. It can be seen from the above equation that bulk modulus has a significant influence on the estimated leakage rate. It is quite difficult to control bulk modulus or to measure the value for each test. A small amount of air in the oil can significantly affect the β value and resulting calculated leakage rates. Because of the poor repeatability of the data and the extreme difficulty in controlling the test parameters, it is recommended that the pressure decay test not be utilized in the revised MERADCOM cylinder specification.

CHAPTER V

PROPOSED CYLINDER TEST PROCEDURES

The current MERADCOM cylinder specification, MIL-C-52762, covers double-acting hydraulic cylinders for use on stationary and mobile equipment. The proposed revision to this specification contains a total of eight separate performance tests which are:

1. Proof Pressure
2. Piston Drift (Position Hold)
3. Packing Drag
4. Cyclic Endurance
5. Corrosion (Salt Spray)
6. Low Temperature
7. Contaminant Wear
8. Wiper Seal Ingression

Details of the test methods including the rationale for their selection and recommended minimum acceptance levels are presented later in this chapter. The actual test procedures which are not current national or international methods are included in Appendix B. Appendix C contains a draft of the performance requirements for the revised specifications.

Proof Pressure

The proof pressure test is included in the current MERADCOM specification and is well accepted as an industry standard method in SAE J214 [3]. The basic difference between the MERADCOM and SAE procedures is the SAE requirement of the pressure held for only 30 seconds whereas the military specification requires 60 seconds. The time does not appear to be a critical parameter; therefore, it is proposed that the SAE procedure be utilized in the revised military specification with a requirement of 30 seconds at a pressure equal to 200% of the operating pressure.

The failure criteria recommended are no evidence of external leakage and no permanent deformation. These criteria are similar to the SAE and consistent with military specification procedures.

Piston Drift

The piston drift test, sometimes called position hold, is also included in the current military specification and in SAE J214. In addition, the SAE committee is currently considering adoption of a method [4] similar to that proposed in Ref. [1]. The piston drift test was selected for inclusion in the revised military specification as a measure of the static sealing capabilities of the piston seal because of the reasons outlined in Chapter IV. The primary test parameters of interest in the piston drift test are the test pressure and test time. The current military specification requires full pressure

and 15 minutes as does SAE J214 which, in addition, requires 50 and 100% pressure checks. The new SAE procedure specifies 5 minutes at 20, 60, and 100% operating pressure.

It is recommended that for the revised military specification the new SAE procedure be utilized for the piston drift test. This test appears to be a compromise between the other tests and includes the latest comments from industry. The industrial survey conducted also revealed that most representatives prefer shorter test times than 15 minutes due to the influences of temperature changes and the failure to obtain additional useful data.

It is suggested that the pass/fail criterion for the piston drift test be based upon the distance traveled by the piston during the five minute period. A maximum rate of travel of 0.4 mm/min or 2 mm during the five minute test period is recommended. This is based upon the requirement of 6.35 mm (.25 inches) in 15 minutes in the current military specification and the industrial survey. Although most industrial respondents disagreed with the exact specification of 6.35 mm in 15 minutes, they were about equally divided about whether it should be higher or lower. The piston drift is an important parameter both from a performance and safety viewpoint; therefore, it is felt that cylinders should at least perform this well.

<u>Packing Drag</u>

The packing drag or no load friction characteristics test is included in the current military specification, in SAE J214, in Ref. [1] and

[2] and is currently processing through the SAE committee as Ref. [5]. This test, although generally considered important, is relatively non-controversial and any of these procedures would suffice. It is recommended that because of its timeliness and industrial acceptance, the new SAE procedure [5] be specified by the revised military specification.

The acceptance criteria specified in the current military specification should be acceptable with the largest portion of industry. These values given in Table 1-1 cover a range of cylinder bores from 22 mm to 360 mm and should be applicable to almost all cylinders covered by the proposed specification.

Cyclic Endurance

Endurance testing of hydraulic components in general is probably one of the most controversial of all test procedures. The first section of this annual report provides a detailed discussion on endurance or durability testing concepts. With cylinder endurance testing, the test time can become quite substantial; therefore, it is important to minimize the test time as much as possible. The current military specification requires an endurance test with a combination of stroke cycling followed sequentially by pressure cycling. The idea behind the pressure cycling is to reduce the test time by utilizing a higher cycle rate during this portion of the test.

The industrial survey conducted resulted in the fact that almost 50% of the respondents use the pressure impulse test frequently and

75% at least occasionally utilize the method. Similarily, a large number of respondents use the stroke cycle test method. Based upon these observations and personal discussions with industrial representatives, it could be concluded that an endurance test sequence similar to that in the current military specification would be acceptable to a large portion of industry.

The primary test parameters disliked by most industrial representatives were the requirement for 20 cycles per minute for the stroke test and 105°C (220°F) during the first portion of the test. Most respondents and previous experience at the Fluid Power Research Center indicate that the stroke cycle should be based upon the stroke or cycle length such that the speed of the piston travel is approximately 15 metres per minute (50 ft/min). Substantially higher cycle speeds may cause abnormal localized heating of the seals and result in premature failure. The elevated temperature during the first portion of the cycle was considered unnecessary and it is suggested that this requirement be eliminated. The test temperature should be changed to 82°C (180°F) for consistency with other test methods. If the above changes were made, it could be concluded that the procedure in MIL-C-52762 would be valid and in line with current state-of-the-art practice.

Another test procedure for evaluating the cyclic endurance or operational integrity of a hydraulic cylinder is now being considered for adoption by the SAE committee. This procedure [6] is based upon methods proposed in Refs. [1] and [2] and utilizes only stroke cycling

and no impulse pressure cycling. One of the advantages of such a test is that the test more closely simulates actual cylinder operation because the piston is cycling. Thus, the piston and rod seals are more subject to wear, flexing, and stress reversals. In addition, only one test fixture and set-up is required. A disadvantage of this type of test is the additional test time required for conducting a stroking test. The proposed SAE procedure, however, specifies a test stroke length requirement of a minimum of only 15% of the full stroke length so that the total test time is generally much less than that required by the current military specification test method. The total distance traveled by the piston during a test conducted in accordance with MIL-C-52762 is twice that traveled with the proposed SAE test; however, if a 30% stroke length is used instead of the minimum 15%, the distance traveled is the same and the proposed SAE test time is still substantially shorter.

Although the proposed SAE operational integrity test appears to be a valid method for determining cyclic endurance, a thorough verification program has not been conducted to illustrate correlation with the current MIL-C-52762 method. Because the SAE method has the potential for substantially reducing the test time, it is recommended that an option be given that either the current MIL-C-52762 procedure modified as stated above or the proposed SAE method be utilized when qualifying under the revised military specification.

The test parameters recommended for use with the proposed SAE technique are the following:

Test Pressure	-	120% rated operating pressure
Test Temperature	-	82°C (180°F)
Rod Velocity	-	15 m/min (50 ft/min)
Total Cycles	-	100% cylinder classification cycles

The test pressure of 120% of rated operating pressure and the test temperature of 82°C were selected for consistency with other endurance test standards (see Section I of this report). The cycle test should be conducted around the midpoint of the test cylinder stroke. For cylinders with cushions, it was recommended that the test be conducted at one end of the cylinder and actual contact be made with the cylinder end on each stroke.

Acceptance criteria for either of the cyclic endurance tests should be no evidence of external leakage except at the rod seal or damage after completion of the designated number of cycles. In addition, it is suggested that external rod seal leakage during either of the stroke cycle tests be limited to a maximum value. This value should be based upon the external dynamic leakage coefficient, K_b, given in Ref. [4] and calculated as follows:

$$K_b = \frac{\text{leakage volume (m}\ell\text{) accumulated in "N" cycles}}{\text{"N" x [sealed circumference (m)]x[2 x stroke length (m)]}}$$

Most respondents who commented on the questionnaire felt that the current specification in MIL-C-52762 of 2 mℓ per 1000 cycles should be lowered and should be based upon the test stroke length. Using a maximum K_b

value of 0.005 as was suggested in Ref. [2], the leakage allowed would be 1 mℓ per 1000 cycles for a typical mobile equipment cylinder with a 5 cm (2 in.) rod and 0.61 m (2 ft.) stroke length. Based upon previous experience and the industrial survey, a K_b value of .005 seems appropriate for most applications.

Finally, the acceptance criteria should include passing the piston drift and packing drag tests conducted upon completion of the required number of test cycles. These tests should be conducted with the cylinder piston centered in the test section in which the cycles were accumulated.

Corrosion

The corrosion or salt spray is included in the current military specification; it was recommended in Refs. [1] and [2], and it is being considered for adoption by the SAE cylinder subcommittee as Ref. [7]. The primary differences between the test specified in MIL-C-52762 and the SAE method are the exposure time and acceptance criteria. The military specification procedure requires exposure to a salt-fog chamber for 30 hours, whereas the SAE is proposing 8 hours. The acceptance criteria in the military specification are no evidence of staining or corrosion that cannot be removed by rubbing with an oily rag and no evidence of pitting. The proposed SAE test requires a rod seal dynamic external leakage be conducted after the exposure period.

Because the test parameters do not appear to be extremely critical in this test, it is recommended that the proposed SAE test procedure be specified in the revised military specification. It is further recommended that the acceptance criteria be specified as no evidence of pitting of the rod surfaces and satisfactory completion of the rod seal dynamic leakage procedure [4] with a maximum external dynamic leakage coefficient, K_b, of 0.005.

Low Temperature

The low temperature test, although it appears in the current version of MIL-C-52762, is not being utilized by a majority of industry. Such a test, therefore, will probably remain to some extent a "military special" requirement. Because military equipment is or can be subjected to extremely cold environments, it is felt that this is still a valid test requirement. Thus, it is suggested that the low temperature test be retained as it exists in MIL-C-52762.

The acceptance criteria recommended for the revised specification are different from the current specification. Currently, it is stated that the cylinder shall operate satisfactorily without piston binding or change in speed of cylinder stroking. This is a rather qualitative indication of cylinder performance. Most industrial representatives responding to the questionnaire felt that a better failure criterion would be specification of maximum external rod leakage at some higher value than under normal conditions. It is, therefore, suggested that a combination of these failure criteria be used such that the cylinder

must operate satisfactorily without piston binding and exhibit a dynamic external leakage coefficient, K_b, at a maximum of 0.025 when tested in accordance with Ref. [4] at full stroke.

Contaminant Wear

The current MIL-C-52762 specification requires an abrasive test which subjects the test cylinder to a dusty environment during cycling. This is a valid test for sensitivity to ingressed contaminant. However, if the cylinder wiper seal is functioning adequately, the amount of ingressed contaminant will be minimal and the abrasive test will not result in contaminant induced wear of the piston seal, cylinder barrel, or rod pressure seal. Previous experience and the majority of industrial respondents indicate that perhaps a more valid approach would be to require a separate wiper seal ingression test and a contaminant wear test to evaluate the sensitivity of the cylinder to fluid-borne contaminants. It is recommended that the revised cylinder specification require these two tests, in order for a cylinder to be qualified.

The cylinder wear test suggested for the revised specification was developed in Refs. [1] and [2] and is given in Appendix B. It includes a requirement for four separate injections of classified AC fine test dust at a 300 mg/ℓ concentration. The primary difference between the procedure given in Ref. [2] and that given in Appendix B is the requirement for a piston drift test after the contaminant injections instead of the dynamic piston leakage test specified in Ref. [2].

The acceptance criteria recommended for use in the revised specification include a maximum rod seal dynamic external leakage coefficient, K_b, of 0.005 during and after the test completion. In addition, a piston drift test conducted after each contaminant injection should result in a rod velocity not higher than 0.4 mm/min.

Wiper Seal Ingression

To complement the contaminant wear test, it is recommended that the SAE J1195 wiper seal ingression test [8] be utilized in the revised MERADCOM specification. This test method evaluates the contaminant ingression characteristics of a cylinder rod wiper seal when subjected to a dusty environment. The method has previously undergone extensive verification testing and is accepted as an SAE standard. Therefore, it should be quite acceptable in the revised cylinder document without major modifications.

Because the standard wiper seal test evaluates the performance of the seal under standard test conditions including rod finish, the test results are independent of the cylinder design actually being considered for approval. It is, therefore, recommended that a requirement be put in the revised cylinder specification that the same rod finish be utilized in the wiper seal test as in the final cylinder design.

The acceptance criterion suggested for the wiper seal test is that the seal must complete 12,220 cycle-metres of operation with the gravimetric level remaining below 140 mg/ℓ. This specification would insure

a high quality seal and the previous verification testing programs have shown that this level is certainly attainable by a large number of seal manufacturers.

General Test Requirements

The current cylinder specification calls for three cylinder classification types as follows:

Type I	700,000 Duty Cycles	(Heavy Duty)
Type II	200,000 Duty Cycles	(Moderate Duty)
Type III	100,000 Duty Cycles	(Light Duty)

Although this classification system may not be optimal and applicable to all applications, there does appear to be a need for such a system. Nearly all equipment manufacturers responding to the survey felt that it is necessary to have cylinder classifications. Very few industrial representatives are using the above classifications exactly; however, their rating systems are certainly not standardized. It is, therefore, suggested that the current classification system be utilized in the revised specification.

Other general test parameters recommended for inclusion in the revised specification, unless otherwise specified in the test procedures, include the following:

Test Fluid	- MIL-L-2104, grade 10
Test Temperature	- $82^{\circ}C \pm 2^{\circ}C$ ($180^{\circ}F$)
Contamination Level	- Not to exceed SAE J1165 solid contaminant code 19/16

These requirements, where different from the current MIL-C-52762, are in accordance with current state-of-the-art technology.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

A complete set of test procedures has been proposed for inclusion in a revised draft of the MERADCOM cylinder specification MIL-C-52762. These procedures and requirements are based upon previous MERADCOM-OSU program conclusions, recent experience, an industrial survey, and participation in industrial standardization meetings. This MERADCOM-OSU program has been aided by the fact that the SAE cylinder sub-committee is currently in the process of revising and adopting many of the test methods previously developed by other MERADCOM-OSU programs. Thus, these procedures will contain the latest state-of-the-art technology.

Some of the proposed SAE standard test methods have not received full SAE approval at the time of this writing. They are proposed for use in the revised cylinder specification, however, because it is felt that any further changes to the documents will be minor in nature and should represent only improvements to the documents.

The two test methods which are felt to require additional verification are the new cyclic endurance test and the contaminant wear test. It is recommended that future effort expended in the area of cylinder

testing be involved with these two test methods. In particular, it is felt that sufficient testing should be conducted to demonstrate correlation between the current MIL-C-52762 cyclic endurance test and the proposed version [6]. Relative to the contaminant wear test, it is suggested that an effort be made to obtain sufficient test data to develop cylinder wear models and life predictions.

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4. "Method for Determining the Leakage Characteristics of A Hydraulic Cylinder," SAE XJXXX Ref. (OSU-HC-1), Draft Copy, Society of Automotive Engineers, 1979.

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APPENDIX A

SUMMARY OF RESULTS FROM INDUSTRIAL
SURVEY ON HYDRAULIC CYLINDER TEST
METHODS AND SPECIFICATION LEVELS

8 February 1979

SUMMARY OF RESULTS FROM INDUSTRIAL SURVEY ON HYDRAULIC CYLINDER TEST METHODS AND SPECIFICATION LEVELS

This questionnaire was conducted as a part of the U.S. Army MERADCOM/Oklahoma State University project to update hydraulic cylinder specification methods. The "basics" of the tests were to be evaluated as well as specifics, such as pass-fail criteria, test temperatures, and speeds. The following is a listing of the questions posed and a summary of the answers received. Several lengthy comments were also submitted and a summary of these will be compiled later.

Number of questionnaires sent: 131

Number of questionnaires completed and returned:
21 classified as cylinder manufacturers (M)
7 classified as users or buyers (U)
10 classified as both manufacturers and users (B)

Hydraulic Cylinder Specification Test Method Evaluation

This survey covers the double-acting hydraulic cylinders for use on stationary and mobile equipment.

1. The following "rating" system is used in current military specifications:

"Expected Service Life"

Type I – 700,000 Duty Cycles (Heavy Duty)
Type II – 200,000 Duty Cycles (Moderate Duty)
Type III – 100,000 Duty Cycles (Light Duty)

1a. Is it necessary to classify cylinders in this way? "Heavy duty," "Moderate duty," etc. by expected service life?

	U	B	M
☐ Yes ☐ No Comments:	7Y	6Y, 4N	8Y, 13N

1b. Does your company/division classify hydraulic cylinders in this way?

	U	B	M
☐ Yes ☐ No	1Y, 6N	5Y, 5N	2Y, 19N

1c. If yes, does your company/division use the *above* ratings exactly?

☐ Yes ☐ No, if not, what ratings are used?

U	B	M
3N, 0Y	1Y, 5N	0Y, 10N

2. Proof pressure: Position and mechanically hold the piston at the approximate midpoint of the cylinder. Fill both sides of the piston with oil. With the cap end port opened, apply an oil pressure equal to twice the working pressure to the rod end of the piston for 60 seconds. With the rod end port opened, apply an oil pressure equal to twice the working pressure to the cap end of the piston for 60 seconds. Any evidence of external leakage or deformation shall constitute failure of this test.

2a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

U	B	M
3F, 2S, 2N	5F, 3S, 2N	10F, 7S, 4N

2b. Is the proof pressure always twice the working pressure?

☐ Yes ☐ No, it is times the working pressure.

U	B	M
2Y, 5N	2Y, 7N	1Y, 19N

2c. A proof pressure test is needed in any procurement specification.

☐ Agree ☐ Disagree

U	B	M
6A, 1D	7A, 3D	17A, 4D

Comments:

3. Piston drift: Position and hold the piston of the cylinder at the approximate midpoint of the cylinder. Fill both sides of cylinder with oil. Cap the head end port and vent the rod end port to atmosphere. Pressurize the head end of the slave cylinder to attain a minimum of specified working pressure at the head end of the test cylinder. Maintain this pressure at an oil temperature of 65°C (150°F), plus or minus 2°C (5°F) for 15 minutes. Measure and record the travel of the piston during this period. Repeat the above test capping rod end port, venting the head end port to atmosphere and pressurizing the rod end of the slave cylinder. Piston drift greater than 0.64 cm (0.25 in.) per 15 minutes shall constitute failure of this test.

3a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

U	B	M
4F, 2S, 1N	6F, 2S, 2N	8F, 8S, 3N

3b. The test temperature of 65°C (150°F) is reasonable and proper.

☐ Yes ☐ No, a temperature of would be superior for such tests.

Comments:

U	B	M
3Y, 4N	4Y, 6N	8Y, 13N

3c. The test time of 15 minutes is reasonable and proper.

☐ Yes ☐ No, a test time of minutes would be better.

Comments:

U	B	M
4Y, 3N	10N	8Y, 12N

3d. It is necessary to measure piston drift in both directions.

☐ Agree ☐ Disagree

U	B	M
6A, 1D	9A, 1D	14A, 7D

3e. Piston drift of 6.35 mm (0.25 inches) per 15 minutes, 0.42 mm (.0167 inches) per minute as the maximum acceptable, is a reasonable requirement for the intended use.

☐ Agree ☐ Disagree, I prefer:

U	B	M
3A, 4D	10D	9A, 11D

3f. A test for piston drift under load is necessary for any cylinder procurement specification.

☐ Agree ☐ Disagree

U	B	M
6A, 1D	7A, 2D	8A, 12D

Comments:

4. Piston and rod packing drag: Position the piston at the approximate midpoint of the cylinder. Fill both sides of the cylinder with oil and vent the head end to atmosphere. Gradually pressurize the rod end of the cylinder. Record the minimum pressure at which the piston moves and also the pressure required to keep it in motion. Repeat the above test by pressurizing the head end of the cylinder. Head end pressure and rod end pressure exceeding the values specified in Table I to actuate the piston shall constitute failure of this test.

Table I. Breakaway Pressure.

Bores, mm (inches) Stroke to 1.8 m (6 feet)	Breakaway Pressure, bar (psi) Rod End	Cap End
25 - 49 (1.0 - 1.9)	2.76 (40)	2.07 (30)
50 - 100 (2.0 - 3.9)	2.41 (35)	1.72 (25)
101 - 200 (4.0 - 7.9)	2.07 (30)	1.38 (20)
201 - 360 (8.0 - 14.0)	1.72 (25)	1.03 (15)

4a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

U	B	M
1F, 5S, 1N	6F, 4S	9F, 5S, 6N

4b. My company/division specifies breakaway pressures that are generally the above.

☐ Higher than ☐ About the same as ☐ Lower than

Comments:

U	B	M
2H, 3A	6H, 2A, 2L	4H, 5A, 5L

4c. Breakaway pressure specifications, such as listed in Table I apply to:

		U	B	M
☐ Oversize and standard rods alike	☐ Any type of packing used	1, 2	4, 5	5, 5
☐ Standard rods only	☐ Standard packings only	2, 3	3, 3	6, 6

4d. Breakaway pressures must be tested in both directions. (Assume rod and bore sizes are known.)

☐ Agree ☐ Disagree

U	B	M
5A, 2D	10A	18A, 2D

4e. Breakaway pressures or some measure of packing drag is needed in any cylinder procurement specification.

☐ Agree ☐ Disagree

U	B	M
4A, 3D	10A	6A, 15D

Comments:

AD-A073 789 OKLAHOMA STATE UNIV STILLWATER FLUID POWER RESEARCH --ETC F/G 13/7

MERADCOM/OSU HYDRAULIC SYSTEM RELIABILITY PROGRAM.(U)

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UNCLASSIFIED OSU-FPRC-9MI NL

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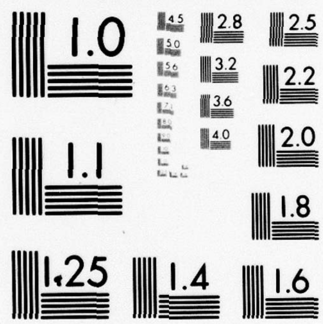

MICROCOPY RESOLUTION TEST CHART

5. Cyclic endurance (stroke cycle): Cycle the test cylinder full stroke, at a minimum of 20 cycles per minute, and at a maximum operating pressure for 30 percent of total number of cycles based on the classification of the cylinder. (See Table I.) Oil temperature shall be 105°C (220°F), plus or minus 2°C (5°F) during the first 10 percent of the stroke cycles, and 65°C (150°F), plus or minus 2°C (5°F) during the remainder of the test run. The pressure, measured at the cylinder ports, shall rise from minimum to specified working pressure in a time interval not to exceed 0.05 seconds. *Measure rod packing total leakage accumulated during entire cycle test.* Upon completion of the above cycling, repeat the piston drift and packing drag tests specified above. Malfunction prior to completion of specified cycles, failure to meet the criteria set forth in above specified tests, or evidence of external leakage or damage shall constitute failure of this test. Rod packing leakage in excess of 2.0 mℓ per 1000 cycles shall constitute failure of this test.

5a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

	U	B	M
	5F, 1S, 1N	5F, 3S, 2N	8F, 5S, 8N

5b. The rate of 20 cycles per minute is reasonable and proper for most applications.

☐ Agree ☐ Disagree

	U	B	M
	2A, 5D	3A, 7D	7A, 13D

5c. Would the specification of maximum and/or minimum cylinder stroking speeds for endurance testing be preferable to the specification of a cycle rate?

☐ Yes ☐ No

	U	B	M
	6Y, 1N	8Y, 2N	15Y, 5N

Comments:

5d. Fifteen meters per second (50 ft/sec) has been suggested as a maximum stroking speed, is this reasonable and proper for endurance testing?

☐ Yes ☐ No

	U	B	M
	3Y, 4N	2Y, 8N	8Y, 9N

Comments:

5e. It is necessary to cycle full stroke in cyclic endurance testing.

☐ Agree ☐ Disagree

	U	B	M
	3A, 4D	5A, 5D	11A, 10D

Comments:

5f. It is my opinion/experience that seals under cycling fail because of:

☐ Reversals in motion	☐ Distance swept	☐ Both	☐ Neither
3M	1U, 3M	3U, 7B, 12M	2U, 2B, 3M

5g. An elevated (105°C, 220°F) oil temperature for the first portion of a cyclic test is necessary and proper.

☐ Agree ☐ Disagree

	U	B	M
	3A, 4D	3A, 6D	3A, 18D

5h. A temperature of 65°C (150°F) is proper for the majority of an endurance test.

☐ Agree ☐ Disagree

	U	B	M
	1A, 6D	2A, 8D	13A, 8D

5i. The pressure rise, from 0 to specified working, in 0.05 seconds is reasonable and proper for this type of test.

☐ Agree ☐ Disagree

U	B	M
6A, 1D	3A, 6D	10A, 11D

5j. My company/division measures rod leakage during cyclic endurance tests.

☐ Frequently ☐ Seldom ☐ Never

U	B	M
4F, 3S	8F, 1S, 1N	18F, 1S, 1N

5k. My company/division measures, after cyclic endurance testing,

	U	B	M
☐ Piston drift under load	6	9	12
☐ Breakaway pressure on packing drag	2	3	7
☐ Other	–	–	–

5l. A maximum acceptable rod leakage rate of 2.0 mℓ per 1000 cycles is reasonable and proper, regardless of stroke length.

☐ Agree ☐ Disagree

U	B	M
2A, 5D	1A, 9D	8A, 10D

6. *Impulse endurance (pressure cycle):* Position the piston rod extended approximately halfway in the test cylinder and pressure cycle alternately the head end and rod end from a minimum pressure to a pressure limited to 125 percent of the working pressure, at a minimum rate of 30 cycles per minute. The pressure, measured at the cylinder ports, shall rise from minimum to a pressure limited to 125 percent of the specified working pressure in a time interval not to exceed 0.05 seconds. Pressure cycle the test cylinder for 70 percent of the total number of cycles based on classification of the cylinder (see 1). Upon completion of above cycling, repeat the piston drift and packing drag tests specified above. Malfunction prior to completion of the specified cycles, failure to meet the criteria set forth in the above specified tests or evidence of external leakage or damage shall constitue failure of this test.

6a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

U	B	M
3F, 2S, 2N	8F, 2N	5F, 9S, 5N

6b. The following parameter values are reasonable and proper for an Impulse Endurance Test.

	U	B	M
Pressure 125% maximum of working — ☐ Agree ☐ Disagree	5A, 1D	3A, 7D	15A, 3D
30 or more cycles per minute — ☐ Agree ☐ Disagree	4A, 2D	9A, 1D	13A, 5D
Pressure rise in 0.05 seconds maximum — ☐ Agree ☐ Disagree	5A, 1D	5A, 4D	12A, 6D

6c. A separate impulse endurance (pressure cycle) test of some form should be required even if stroke cycle testing is done.

☐ Agree ☐ Disagree

U	B	M
3A, 3D	6A, 4D	12A, 7D

6d. It is necessary to rerun the following tests after impulse endurance testing.

4U, 7B, 9M	1U, 4M	
☐ Piston drift	☐ Packing drag	☐ Other (specify)

Comments:

7. *Salt spray test:* The piston rod shall be subjected to a 5 percent NaCl solution, salt-spray test in accordance with ASTM B-117 for a period of not less than 30 hours. Examine the piston rod surfaces for corrosion at conclusion of the test. Staining or corrosion that cannot be removed by rubbing with an oily rag or evidence of pitting shall constitute failure of this test.

7a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

U	B	M
2F, 2S, 3N	3F, 3S, 4N	3F, 9S, 9N

7b. The above test is necessary for most procurement specifications.

☐ Agree ☐ Disagree

U	B	M
4A, 2D	8A, 2D	4A, 15D

Comments:

8. *Abrasive test:* The piston rod shall be subjected to a dust test in accordance with MIL-STD-810, Test Method 510 (controlled atmosphere dust box). The piston rod shall be cycled for 10 hours at rate of 20 cycles per minute. At the conclusion of the test, examine the piston rod for wear and damage. Upon the completion of this test, repeat the packing drag test specified herein. Upon completion of the drag test, cycle the cylinder full stroke for 1000 cycles. *Measure and record the rod packing leakage.* Failure to meet the criteria set forth in the above specified tests shall constitute failure of this test.

8a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

U	B	M
1F, 2S, 4N	1F, 2S, 7N	5S, 16N

8b. This test is necessary and useful even if specific wiper seal test data is available.

☐ Agree ☐ Disagree

U	B	M
3A, 4D	1A, 8D	3A, 13D

8c. The dust environment specified is proper and reasonable for use with this test.

☐ Agree ☐ Disagree

U	B	M
4A, 1D	1A, 3D	7A, 8D

8d. It is necessary that the following be evaluated after testing.

6U, 2B, 11M	3U, 1B, 4M	7U, 5B, 12M	
☐ Rod condition	☐ Packing drag	☐ Rod leakage	☐ Other (specifiy)

☐ None

8e. The above test, or similar, is necessary for use in procurement specifications.

☐ Agree ☐ Disagree

U	B	M
3A, 5D	3A, 5D	3A, 14D

Comments:

8f. Could a wiper seal test, such as SAE J1195 replace the *Abrasive Test*?

☐ Yes ☐ No

U	B	M
4Y, 1N	4Y	12Y, 1N

Comments:

9. *Low temperature:* Unless otherwise specified, the cylinder shall operate at minus 32°C (-25°F) after storage at minus 35°C (-30°F) without piston binding or change in speed of cylinder stroking. When specified, the cylinder shall operate satisfactorily at minus 46°C (-50°F) after storage at the same temperature without piston binding or change in speed of cylinder stroking.

9a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

U	B	M
1F, 3S, 3N	3S, 5N	4S, 16N

9b. The above test, or similar, should be included in a procurement specification.

☐ Agree ☐ Disagree

U	B	M
4A, 3D	3A, 5D	4A, 13D

9c. Is "change in stroking speed" a valid failure criterion?

☐ Yes ☐ No, I prefer:

U	B	M
1Y, 4N	3Y, 3N	1Y, 14N

There are a number of test procedures for double acting hydraulic cylinders not presently included in military specifications.

10. Static Internal Leakage, Pressure decay technique: The piston rod is blocked in midstroke. Pressure is applied to one end of the cylinder. The pressurized volume is sealed off, trapping the pressure. The rate of pressure decay is measured; then, using the Bulk modulus and the trapped cylinder volume, the rate of internal leakage is determined.

10a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

U	B	M
1F, 2S, 4N	2F, 2S, 7N	1F, 5F, 14N

10b. The bulk modulus of the test oil will be constant or can be controlled and measured from test to test.

☐ Agree ☐ Disagree

U	B	M
1A, 6D	5A, 4D	4A, 7D

11. Contaminant Wear: The hydraulic cylinder is operated with oil contaminated to 300 mg/ℓ of AC Fine Test Dust. The cylinder is cycled under load for a specified number of cycles. Then a piston drift test is conducted.

11a. My company/division uses the above test, or similar.

☐ Frequently ☐ Seldom ☐ Never

U	B	M
2S, 5N	1S, 9N	2S, 18N

11b. My company/division is concerned with cylinder degradation under contaminated conditions.

☐ Yes ☐ No

U	B	M
7Y	8Y, 2N	15Y, 3N

11c. The following tests should be run following the contaminant test above.

2U, 6B, 10M	2U, 4M	5U, 1B, 7M	
☐ Piston Drift	☐ Packing Drag	☐ Dynamic Leakage (internal)	☐ Other

11d. The 300 mg/ℓ, ACFTD, is a reasonable and proper contamination level.

☐ Agree ☐ Disagree

U	B	M
5A, 1D	3A, 5D	8A, 3D

11e. The current plans utilize full distribution ACFTD only. Is this acceptable or are classified dust tests necessary?

☐ Acceptable ☐ Need cut-dust, why?

U	B	M
6A, 1N	8A, 1N	8A, 1N

11f. Is this test necessary if a wiper seal (dust box) test is also conducted?

☐ **Yes** ☐ **No**

	U	B	M
	3Y, 3N	4Y, 4N	7Y, 9N

Comments:

12. Cushioning: Velocity versus rod displacement plots are presently used to evaluate hydraulic cylinder cushions. Forces used result in 1.0, 1.5 and 2.0 times the rated pressure within the cylinder.

12a. My company/division uses the above test, or similar.

☐ **Frequently** ☐ **Seldom** ☐ **Never**

	U	B	M
	2S, 4N	4S, 4N	4F, 6S, 10N

12b. My company/division uses/manufactures the following proportions of cylinders.

Fixed cushions	________%	≈ 5%
Adjustable cushions	________%	≈ 5%
No cushions	________%	≈ 90%

12c. Is a standard hydraulic cylinder cushion test method needed?

☐ **Yes** ☐ **No**

	U	B	M
	3Y, 3N	5Y, 5N	11Y, 6N

Comments:

12d. Is measurement of velocity versus displacement a valid technique for evaluating cylinder cushions?

☐ **Yes** ☐ **No**

	U	B	M
	4Y, 1N	6Y, 3N	10Y, 4N

13. Dynamic leakage, direct measurement: One port of the cylinder is opened and drained. The opposite port is pressurized and the cylinder is stroke cycled. Leakage from the open port is measured to determine dynamic leakage rate.

13a. My company/division uses the above test, or similar

☐ **Frequently** ☐ **Seldom** ☐ **Never**

	U	B	M
	2F, 2S, 3N	2F, 1S, 7N	6F, 3S, 11N

13b. Dynamic leakage rate is an important cylinder parameter.

☐ **Yes** ☐ **No**

	U	B	M
	6Y, 1N	1Y, 7N	4Y, 16N

13c. The life and operation of the piston seal would not be affected by one side being run "dry".

☐ **Agree** ☐ **Disagree**

	U	B	M
	1A, 6D	10D	5A, 15D

APPENDIX B

CYLINDER TEST METHODS

SAE/OSU-HC-1

SAE/OSU-HC-2

SAE/OSU-HC-4

SAE/OSU-HC-5

OSU-HC-6

METHOD FOR DETERMINING THE LEAKAGE CHARACTERISTICS OF A HYDRAULIC CYLINDER

INTRODUCTION - In hydraulic fluid power systems, a hydraulic cylinder is a means to convert fluid pressure into mechanical force. The configuration and materials used in the design of a cylinder are critical factors influencing its function. The internal and external leakages of a cylinder are a relative indication of the performance capabilities of that cylinder.

1.0 SCOPE - To provide a laboratory method for determining the internal and external leakage of a hydraulic cylinder.

2.0 PURPOSE - To verify the capability of a cylinder to seal fluid pressure under specified cycling or holding conditions.

3.0 DEFINITIONS

3.1 CYCLE - One extension and retraction of the cylinder rod for a specified stroke length.

3.2 RATED PRESSURE - Operating pressure as specified by the manufacturer. This pressure is, normally, the pressure at which the manufacturer recommends continuous duty.

3.3 STROKING LENGTH - The distance travelled by the piston in completing one-half cycle.

3.4 INTERNAL SEAL - A seal or seal set configuration which restricts leakage flow when pressurized fluid is applied on either side of piston.

3.5 EXTERNAL SEAL - A seal configuration which restricts leakage flow to the outside of a cylinder when pressurized fluid is applied.

3.6 CYCLE RATE - The number of cycles per unit of time.

3.7 DYNAMIC LEAKAGE - The volume of fluid leaking past a seal under cycling conditions.

3.8 "N" - The number of cycles measured to determine leakage coefficients, K_a and K_b.

3.9 DRIFT - A result of internal static leakage and an indication of the load holding ability of a cylinder, measured by recording rod movement (drift) during a specified time interval while resisting an applied load.

4.0 TEST CONDITIONS

4.1 TEMPERATURE - The temperature of the fluid for this test shall be specified by the cylinder manufacturer. Ambient Temperature during this test shall be 20°C (68°F) minimum.

4.2 PRESSURE - The internal cylinder pressure shall be equal to the manufacturer's rated pressure or other pressure as agreed upon between user and supplier.

4.3 CYCLE RATE - The cycle rate shall be specified by the manufacturer.

4.4 CONTAMINATION LEVEL - The contamination level shall not exceed SAE J1165 solid contaminant code 19/16.

4.5 ACCURACY OF MEASUREMENT - The accuracy of measurement for all test parameters shall be specified by the testing laboratory and reported as part of the test data.

5.0 TEST EQUIPMENT

5.1 Use a suitable test fixture, for example, an oscillating-beam type, a conventional in-line beam type, or a similar fixture per circuit schematic - Figure B-1.

5.2 Use a suitable control filter capable of maintaining the required fluid cleanliness level.

5.3 Test fluid shall be specified by the manufacturer.

6.0 PRELIMINARY PROCEDURES

6.1 Circulate fluid and cycle the piston rod of the test cylinder until the required cleanliness level has been reached.

7.0 TEST PROCEDURE - INTERNAL SEAL

7.1 DYNAMIC LEAKAGE TEST - Caution. Some seals can be damaged during the dynamic leakage test because it possibly results in the seal running dryer than normal. Consideration should be given to seal material and construction before using this dynamic leakage test.

7.1.1 Disconnect return line from test cylinder and attach to load cylinder as shown in Figure B-2.

7.1.2 Resume cycling at rated temperature and adjust valving to obtain rated pressure on the test cylinder.

7.1.3 Continue cycling test cylinder for a minimum of 100 cycles, then measure and record any leakage for "N" complete cycles of the moving element.

7.1.4 Using the leakage volume obtained in 7.1.3 calculate the internal dynamic leakage coefficient, K_a, according to the following expression:

$$K_a = \frac{\text{Leakage Volume (m}\ell\text{) Accumulated in "N" Cycles}}{\text{"N" [Sealed Circumference (m)] [(2 x Stroke Length (m)]}}$$

7.1.5 Repeat the above steps, for leakage from the rod end to the cap end, by revising connections. (This step may be omitted if agreed upon by the parties concerned).

7.2 DRIFT TEST

7.2.1 Cycle cylinder until the fluid temperature is stabilized. Stop cylinder at mid stroke.

7.2.2 Insert a pressure gauge in pressure port at cap end of cylinder. Leave rod end open to atmosphere.

7.2.3 Apply external load to cylinder sufficient to produce 20% of rated pressure at cap end of cylinder. Repeat at 60% and 100% of rated pressure or as agreed upon.

7.2.4 Hold at specified pressure and record rod travel. Read and record dial indicator for 5 minutes at one minute intervals. Maintain temperature of cylinder within $2^{o}F$ ($1^{o}C$) or errors due to oil volume change could result.

7.2.5 Repeat test at rod end of cylinder (See 7.2.3).

8.0 TEST PROCEDURE - EXTERNAL SEAL

8.1 Cycle the moving element at rated pressure, temperature, and speed.

8.2 Continue cycling test cylinder for a minimum of 100 cycles, then measure and record the accumulated leakage for "N" complete cycles of the moving element.

8.3 Calculate the external dynamic leakage coefficient using the following expression:

$$K_b = \frac{\text{Leakage Volume (m}\ell\text{) Accumulated in "N" Cycles}}{\text{"N" [Sealed Circumference (m)] [2 x Stroke Length (m)]}}$$

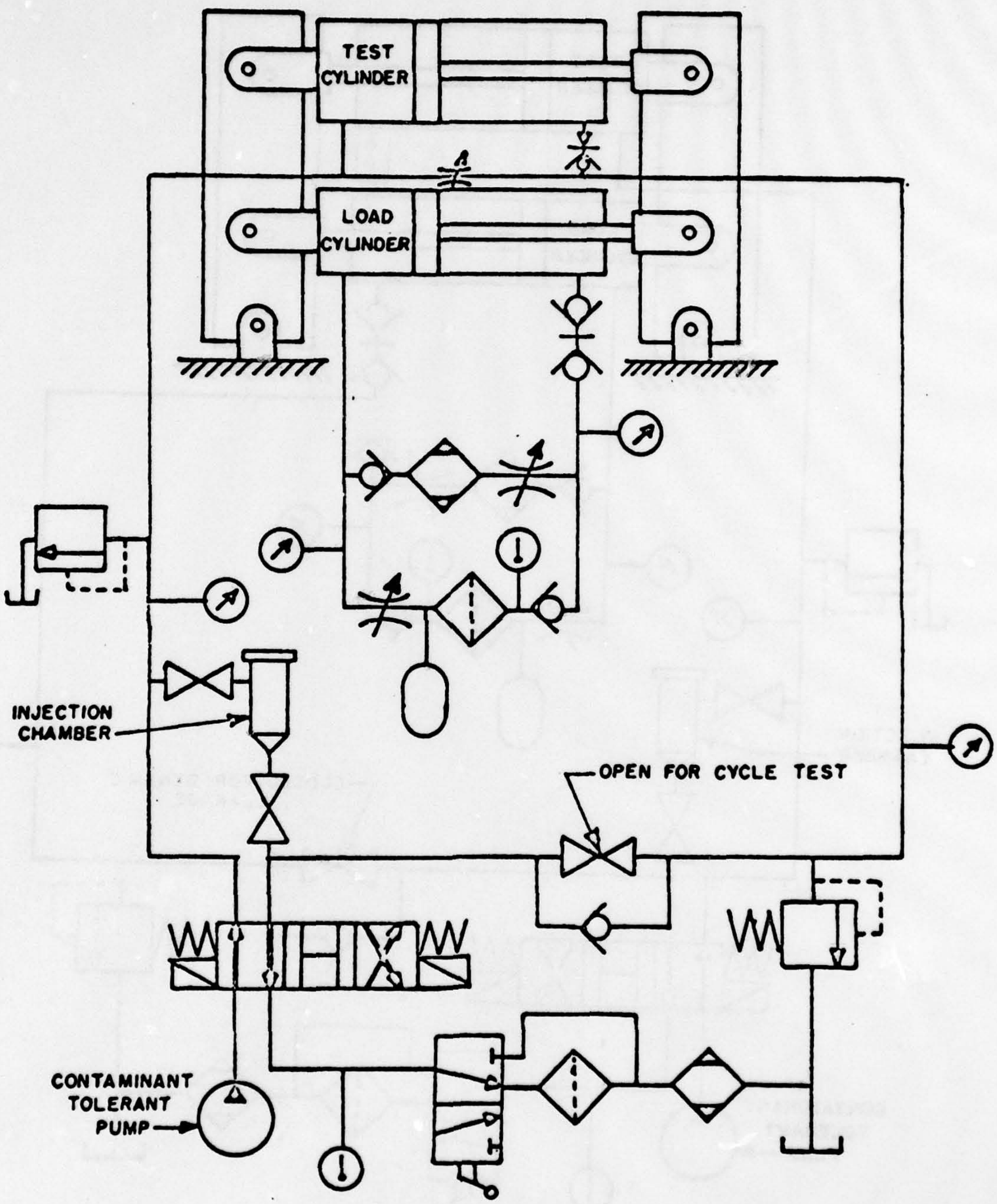

Fig. B-1 Typical Cylinder Test Circuit Schematic (Cycling).

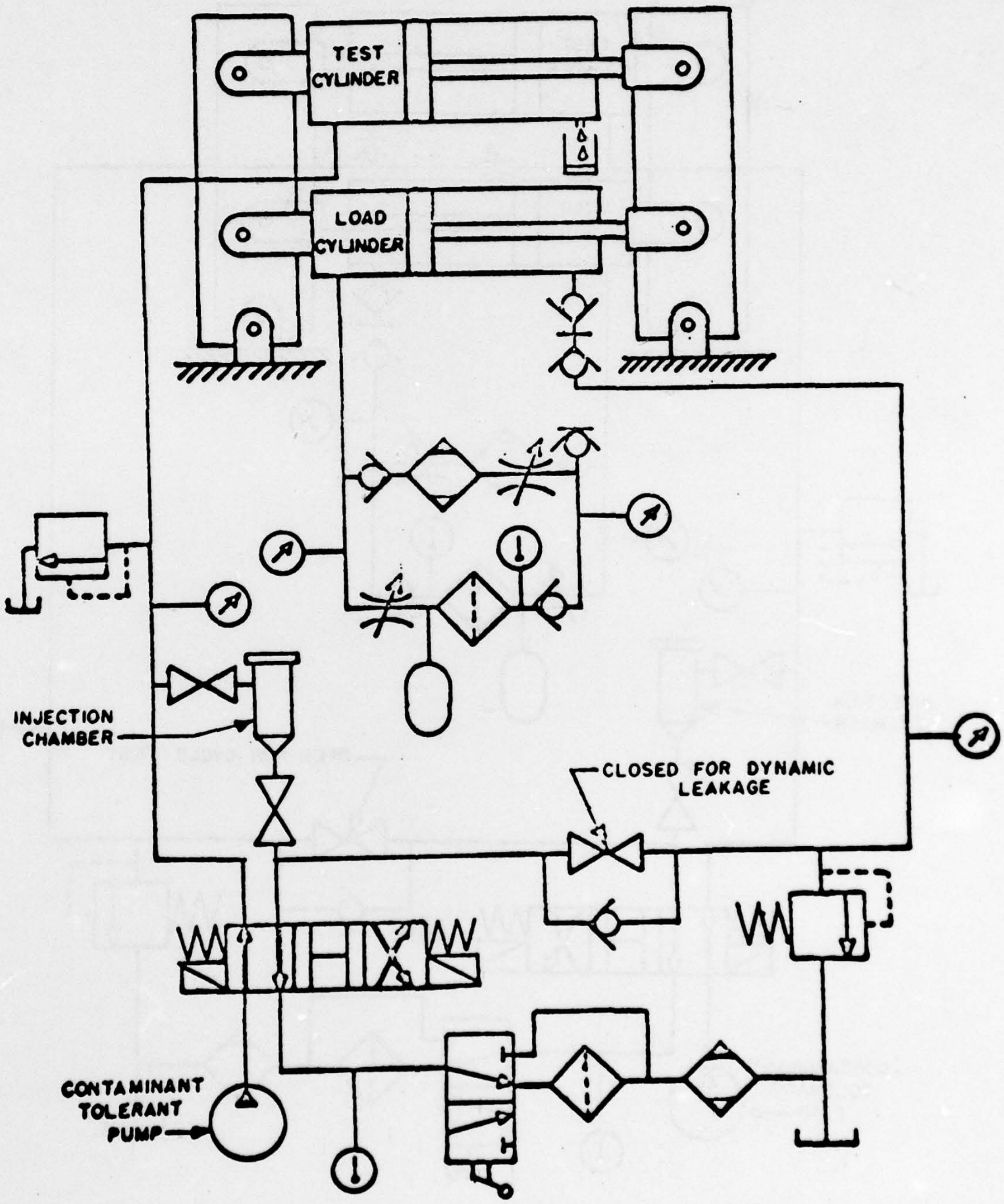

Fig. B-2 Typical Cylinder Test Cylinder Schematic (Dynamic Leakage).

METHOD FOR DETERMINING THE NO LOAD FRICTION CHARACTERISTICS OF A HYDRAULIC CYLINDER

INTRODUCTION - In hydraulic fluid power systems, a hydraulic cylinder is a means to convert fluid pressure into mechanical force. The configuration and materials used in the design of a cylinder are critical factors influencing its function. The static and dynamic friction characteristics of a hydraulic cylinder are an indication of its mechanical efficiency.

1.0 SCOPE - To provide a laboratory method for determining the static and dynamic friction associated with a hydraulic cylinder.

2.0 PURPOSE - To verify the static and dynamic friction of a hydraulic cylinder under no external load.

3.0 DEFINITIONS

3.1 STATIC BREAKAWAY PRESSURE - The minimum pressure which initiates movement of the piston under no external load condition.

3.2 DYNAMIC DRAG PRESSURE - The minimum pressure required to maintain motion after breakaway.

4.0 TEST CONDITIONS

4.1 Test fluid as specified by the cylinder manufacturer.

4.2 Test temperature as specified by the cylinder manufacturer (usually) at room temperature).

4.3 The accuracy of measurement for all test parameters shall be as specified by the testing laboratory and reported as part of the test data.

5.0 TEST EQUIPMENT

5.1 Use a fluid power source capable of delivering a pressure necessary to overcome the static breakaway pressure of the cylinder.

5.2 Use a means of indicating pressure at the rod end and cap ports.

6.0 TEST PROCEDURE

6.1 Position the piston at the midpoint of the cylinder under test.

6.2 Fill both sides of the cylinder with hydraulic fluid, install pressure measuring instrumentation in the rod end port and vent the cap end to atmosphere.

6.3 After allowing a minimum of two minutes to permit the rod and piston seals to settle in normal static positions, apply gradually increasing pressure to the rod end of the cylinder under test.

6.4 Record on data sheet the static breakaway pressure, the dynamic drag pressure and the flow rate.

6.5 Repeat the above test for the reverse direction by installing pressure-measuring instrumentation in the cap end port and applying pressure to the cap end port and venting the rod end to atmosphere.

6.6 Record on data sheet the static breakaway pressure, the dynamic drag pressure, and the flow rate.

METHOD FOR VERIFYING THE OPERATIONAL INTEGRITY OF A HYDRAULIC CYLINDER

INTRODUCTION - In hydraulic fluid power systems, a hydraulic cylinder is a means to convert fluid pressure into mechanical force. The configuration and materials used in the design of a cylinder are critical factors influencing its function. The operational integrity of a hydraulic cylinder operating as an integral part of a structural linkage is an important parameter.

1.0 SCOPE - To provide a laboratory method for determining the operational integrity of a hydraulic cylinder.

2.0 PURPOSE - To verify the operational integrity of a hydraulic cylinder by a proof pressure and an operational test.

3.0 DEFINITIONS

3.1 PROOF PRESSURE - The maximum pressure which can be applied that will not cause any permanent set or deformation. Normally this is at least 200% of rated pressure.

3.2 FIRST FAILURE LIFE - The number of pressure cycles a cylinder can withstand and not fail to function as specified.

3.3 CYCLE - One extension and retraction of the cylinder rod for a specified stroke length.

3.4 CYCLE RATE - The number of cycles per unit of time.

3.5 RATED PRESSURE - Operating pressure as specified by the manufacturer. This pressure is, normally, the pressure at which the manufacturer recommends continuous duty.

3.6 STROKING LENGTH - The total distance traveled by the piston in completing one-half cycle.

4.0 TEST CONDITIONS

4.1 Fluid test temperature measured in supply line, shall be 50°C (122°F) and/or 90°C (194°F) or as agreed between user and supplier.

4.2 Operational test pressure shall the manufacturer's rated pressure and measured at the cylinder work ports. A 10 percent transient overshoot is permissable.

4.3 PRESSURE RISE RATE - Shall be a minimum of 1000 bars (14500 PSI) per second.

4.4 CONTAMINATION LEVEL - Test system shall have a contamination level not to exceed SAE J1165 solid contaminant code 19/16.

4.5 STROKING LENGTH - The length of stroke for the operational test shall be equal to at least 15 percent of the maximum stroke length of the hydraulic cylinder.

4.6 CYCLE RATE - As specified by the manufacturer.

4.7 FIRST FAILURE LIFE RATING - As specified by the manufacturer.

4.8 PROOF PRESSURE - As specified by the manufacturer.

4.9 ACCURACY OF MEASUREMENT - The accuracy of measurement for all test parameters shall be specified by the testing laboratory and reported as part of the test data.

5.0 TEST EQUIPMENT

5.1 Use a suitable test fixture, for example, an oscillating-beam type, a conventional in-line beam type, or similar fixture per circuit schematic - Figure 3-2..

6.0 TEST PROCEDURES

6.1 Connect test cylinder to a fluid power source and cycle maximum stroke a minimum of 20 cycles to displace the trapped air.

6.2 Extend the cylinder rod and pressurize the cap port to the specified proof pressure. Hold one minute minimum.

6.3 Retract the cylinder rod and pressure the rod end port to the specified proof pressure. Hold one minute minimum.

6.4 Install temperature-measuring instrumentation on the outside barrel wall at the point corresponding to one-half the stroking length used in this test.

6.5 Install test cylinder in cycle test stand.

6.6 Position the cylinder rod to cycle in the middle section of the stroke.

6.7 Cycle test cylinder for one hundred cycles at rated pressure to simulate a break-in period.

6.8 Conduct leakage tests per SAE XJXXX REF. (OSU-HC-1) on both internal and external seals.

6.9 Cycle the cylinder against the external load at the specified operational test pressure and cycle rate.

6.10 Continue the specified cycle until failure or until first failure life rating is reached, 700,000, 200,000 or 100,000 duty cycles, or as specified. Measure seal leakage every 50,000 cycles, or at agreed upon intervals per SAE XJXXX REF. (OSU-HC-1).

6.11 No modifications or repairs shall be made during the operational test.

6.12 Pin Joints shall be greased when necessary.

7.0 <u>PRESENTATION OF RESULTS</u>

7.1 Calculate and report the dynamic seal leakage coefficients at all measured points according to SAE XJXXX REF. (OSU-HC-1).

7.2 Plot graphs of seal leakage coefficients versus number of cycles.

7.3 Report any failures of external leakage observed.

METHOD FOR VERIFYING THE CORROSION RESISTANCE CAPABILITY OF A HYDRAULIC CYLINDER

INTRODUCTION - In hydraulic fluid power systems, a hydraulic cylinder is a means to convert fluid pressure into mechanical force. The configuration and materials used in the design of a cylinder are critical factors influencing its function. The capability of a cylinder to resist atmospheric corrosion is an important parameter in its application.

1.0 SCOPE - To provide a laboratory method for determining the capability of a hydraulic cylinder to withstand corrosion.

2.0 PURPOSE - To provide an accelerated method to verify the corrosion-resistance characteristics of a hydraulic cylinder.

3.0 DEFINITIONS

3.1 CYCLE - One extension and retraction of the cylinder rod for a specified stroke length.

3.2 RATED PRESSURE - Operating pressure as specified by the manufacturer. This pressure is, normally, the pressure at which the manufacturer recommends continuous duty.

3.3 CYCLE RATE - The number of cycles per unit of time.

4.0 TEST CONDITIONS

4.1 Test temperature as specified by the manufacturer.

4.2 Salt-fog environment as specified in ASTM-B-117.

4.3 Test pressure is manufacturer's rated pressure.

4.4 Cycle rate as specified by the manufacturer.

5.0 TEST EQUIPMENT

5.1 Use a salt-fog chamber having an atomizer and an air circulating system.

5.2 Use a suitable test fixture, for example, an oscillating-beam type, a conventional in-line beam type, or a similar fixture per circuit schematic - Figure C-2.

6.0 TEST PROCEDURE

6.1 Install temperature measuring instrumentation into fluid stream at supply port to measure inlet fluid temperature.

6.2 Install the cylinder in the test fixture.

6.3 Cycle the cylinder against the external load for 1000 cycles and perform a 1000 cycle rod seal dynamic leakage test per SAE XJXXX REF. (OSU-HC-1).

6.4 Remove the cylinder from the test fixture and place the fully extended cylinder in the salt-fog chamber, clean and remove any trace of oil film from the rod, and expose the cylinder to the specified salt-fog environment for 8 hours.

6.5 Remove the cylinder from the salt-fog chamber and install it in the test fixture.

6.6 Cycle the cylinder against an external load at rated conditions for 10,000 full-stroke cycles.

6.7 Perform a rod seal dynamic leakage test per SAE XJXXX REF. (OSU-HC-1) during the final 1000 cycles.

7.0 PRESENTATION OF RESULTS

7.1 Calculate and report the rod seal dynamic leakage coefficients, K_b, per paragraph 8.3 of SAE XJXXX REF. (OSU-HC-1). Use the leakage values obtained in Steps 6.3 and 6.7.

METHOD FOR DETERMINING THE CONTAMINANT SENSITIVITY OF A HYDRAULIC CYLINDER

INTRODUCTION - The useful life of a hydraulic cylinder is directly related to its assembled configuration, fabrication materials, hydraulic fluid, and its operating conditions. The service life of a cylinder is considered to be terminated when sufficient wear has occurred that it no longer can hold or maintain a given position under load conditions. The rate of wear within hydraulic cylinders is proportional to the contamination level of the hydraulic fluid exposed to the internal surfaces of the cylinder. The contaminant sensitivity of a hydraulic cylinder is a direct reflection of the contaminant wear which might be expected in service.

1.0 SCOPE - To provide a method for determining the contaminant sensitivity of a hydraulic cylinder. Degradation by processes other than contaminant wear is not considered.

2.0 PURPOSE - To verify the contaminant sensitivity of a hydraulic cylinder at the operating conditions specified by the cylinder manufacturer.

3.0 DEFINITIONS

3.1 CONTAMINANT SENSITIVITY - The increase in rod and piston seal dynamic leakage coefficients with respect to fluid particulate contamination.

3.2 CYCLE - One full extension and one full retraction stroke.

3.3 STROKING LENGTH - The distance traveled by the piston while completing one-half cycle.

4.0 TEST CONDITIONS

4.1 TEST TEMPERATURE - Test oil temperature shall be as specified by the manufacturer.

4.2 TEST PRESSURE - Test pressure shall be the manufacturer's rated pressure.

4.3 CYCLE RATE - The time required to make one complete cycle shall be specified by the manufacturer.

4.4 INJECTION GRAVIMETRIC LEVEL - 300 mg/litre

4.5 SIZE RANGE INJECTIONS OF CLASSIFIED AC FINE DUST

1. 0-20 micrometres
2. 0-40 micrometres
3. 0-60 micrometres
4. 0-80 micrometres

4.6 CLEANLINESS LEVEL - The fluid shall be filtered prior to each contaminant injection to less than SAE J1165 Solid Contaminant Code 19/16.

4.7 CONTAMINATED FLUID VOLUME - The volume of the test cylinder circuit less the filter system shall be equal to five times the extended test cylinder volume.

4.8 ACCURACY OF MEASUREMENT - Accuracy of measurements shall be specified by the manufacturer and reported as part of the test data.

4.9 STROKING LENGTH - The stroking length during this test shall be at least 50 percent of the total available stroke. The actual stroking length used shall be reported.

5.0 TEST EQUIPMENT

5.1 Use a cylinder test fixture per Figure B-5 capable of cycling the test cylinder with an external load.

5.2 Use a test circuit with means to inject contaminant into the test cylinder circuit.

5.3 Use a test circuit with means to extract representative fluid samples from the test cylinder circuit.

5.4 Lines in the test cylinder circuit shall be sized to insure turbulent mixing conditions.

5.5 Use a hydraulic pump and directional valve with minimum sensitivity to contaminant wear.

5.6 Use a control filter capable of maintaining the required cleanliness level.

5.7 Use a test circuit with a by-pass valve installed around the test cylinder in order to insure circulation of the contaminated fluid.

5.8 The test fixture shall be capable of imposing both axial and lateral external loads on the piston and piston rod of the test cylinder. Either an oscillating beam fixture which allows rotating-pin forces and full-cylinder weight to be imposed on the test cylinder or a conventional-type beam fixture can be used. In the conventional in-line fixture, a means shall be provided for applying a side load (lateral) at the test cylinder bearing equal to 10 percent of the output force resulting from rated pressure.

5.9 Test oil shall be specified by the manufacturer.

6.0 <u>TEST PROCEDURE</u>

6.1 Install the test cylinder into the test fixture with ports in the down position and adjust the test cylinder circuit volume as specified.

6.2 Cycle the piston rod of the test cylinder at the specified cycle rate and at a maximum of 25 percent of rated pressure while circulating the test fluid through the control filter. Continue cycling until the required cleanliness level has been reached.

6.3 Perform a piston drift test and a rod dynamic leakage test per SAE XJXXX REF. (OSU-HC-1).

6.4 Block the control filter from the test circuit.

6.5 Continue cycling at the specified rate and 25 percent of rated pressure.

6.6 Inject the proper quantity of 0-20 micrometre size range contaminant into the test circuit to create a 300 mg/litre concentration.

6.7 Continue cycling for 200 cycles at 25 percent rated pressure, then block injection chamber from test circuit.

6.8 Increase the cycling pressure to 100 percent rated.

6.9 Continue cycling for 1000 cycles.

6.10 Repeat Steps 6.2 through 6.9 for each of the specified size range injections.

6.11 Repeat Steps 6.2 and 6.3.

7.0 <u>PRESENTATION OF RESULTS</u>

7.1 Record cylinder identification, operating conditions, and test data.

7.2 Plot the rod dynamic external leakage coefficients, K_b, and the piston drift values versus upper size of the injection ranges.

APPENDIX C

PROPOSED DRAFT OF

MILITARY SPECIFICATION

CYLINDER, HYDRAULIC, DOUBLE-ACTING

PROPOSED DRAFT OF MILITARY SPECIFICATION

CYLINDER, HYDRAULIC, DOUBLE-ACTING

1.0 SCOPE

1.1 SCOPE - This specification covers the double-acting hydraulic cylinders for use on stationary and mobile equipment.

1.2 CLASSIFICATION - Hydraulic cylinders shall be of the following types as specified:

TYPE I	700,000 Duty Cycles (Heavy Duty)
TYPE II	200,000 Duty Cycles (Moderate Duty)
TYPE III	100,000 Duty Cycles (Light Duty)

2.0 APPLICABLE DOCUMENTS

2.1 TEST PROCEDURES - This specification requires the use of test procedures adopted by the Society of Automotive Engineers and proposed by Oklahoma State University.

3.0 REQUIREMENTS

3.1 The manufacturer shall specify the maximum rated flow and the cylinder duty cycle classification for all cylinders submitted.

3.2 PERFORMANCE - The cylinder shall satisfy the performance requirements specified herein when operating at the manufacturer's maximum rated flow or cycle rate, which result in a minimum average

rod speed of 15 m/min (50 ft/min) during extension and retraction.

3.3 PROOF PRESSURE - When tested in accordance with SAE J214 at 200% operating pressure for 30 seconds, the cylinder shall exhibit no evidence of external leakage and no permanent deformation.

3.4 PISTON DRIFT - When tested in accordance with the SAE XJXXX REF. (OSU-HC-1) piston drift test, the piston drift shall not exceed 2.0 mm during the 5 minute test in either the extend or retract directions.

3.5 PACKING DRAG - When tested in accordance with SAE XJXXX REF. (OSU-HC-2), the maximum pressure required to actuate the piston shall not exceed the value shown in the following table:

Bores, mm (inches) Stroke to 1.8m (6 feet)	Breakaway Pressure, bar (psi) Rod End	Cap End
25-49 (1.0 - 1.9)	2.76 (40)	2.07 (30)
50-100 (2.0 - 3.9)	2.41 (35)	1.72 (25)
101-200 (4.0 - 7.9)	2.07 (30)	1.38 (20)
201-360 (8.0 - 14.0)	1.72 (25)	1.03 (15)

3.6 CYCLIC ENDURANCE

3.6.1 OPTION I - Test in accordance with the cyclic and impulse endurance tests in MIL-C-52762 with the exception of the use of a 15m/-min cycle speed and a test temperature of 82°C.

3.6.2 OPTION II - Test in accordance with the SAE XJXXX REF. (OSU-HC-4) operational integrity method with a test pressure of 120% operating pressure, a test temperature of 82°C, a rod velocity of 15m/-min, and a total number of test cycles equal to 100% of the cylinder classification duty cycles.

3.6.3 ACCEPTANCE CRITERION - Upon completion of the test, the cylinder shall exhibit no evidence of external leakage except at the rod seal; no physical damage; a rod seal external leakage coefficient, K_b, no greater than 0.005 when tested in accordance with SAE XJXXX REF. (OSU-HC-1); and shall pass the piston drift and packing drag tests per 3.4 and 3.5

3.7 CORROSION RESISTANCE - When tested in accordance with SAE XJXXX REF. (OSU-HC-5), the cylinder rod surface shall exhibit no evidence of pitting and the cylinder shall satisfactorily complete a rod seal dynamic leakage test per SAE XJXXX REF. (OSU-HC-1) with a maximum allowable external leakage coefficient, K_b, of 0.005.

3.8 LOW TEMPERATURE - Unless otherwise specified, the cylinder shall operate at minus 32°C after storage, at minus 34°C without piston binding, and exhibit a dynamic external leakage coefficient, K_b, not greater than 0.025 when tested in accordance with SAE XJXXX REF. (OSU-HC-1). When specified, the cylinder shall be tested at minus 46°C with the above acceptance criteria.

3.9 CONTAMINANT WEAR - After testing in accordance with OSU-HC-6, the cylinder shall be subjected to piston drift (see 3.4) and exhibit a piston drift not to exceed 3.0 mm during the 5 minute period. In addition, the rod seal external leakage coefficient, K_b, shall not exceed 0.005 when tested per SAE XJXXX REF. (OSU-HC-1).

3.10 WIPER SEAL INGRESSION - When the cylinder rod wiper seal is tested in accordance with SAE J1195 using a rod finish as in the final cylinder design, the seal must complete 12,200 cycle-metres of operation with the gravimetric level remaining below 140mg/ℓ.

3.11 TEST SEQUENCE - Two cylinders are required for the tests covered by this specification and are to be tested in the following sequence:

SAMPLE NO. 1	SAMPLE NO. 2
1. Proof Pressure	1. Corrosion Resistance
2. Piston Drift	2. Contaminant Wear
3. Packing Drag	
4. Low Temperature	
5. Cyclic Endurance	

In addition, a rod wiper seal and cylinder rod are required for the wiper seal ingression test.

MERADCOM/OSU HYDRAULIC SYSTEM RELIABILITY PROGRAM

SECTION III

HYDRAULIC VALVE SPECIFICATIONS

PREPARED BY PERSONNEL OF

FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA

May 1979

FINAL REPORT

June 1978 – May 1979

PREPARED FOR

U.S. ARMY MOBILITY EQUIPMENT RESEARCH
AND DEVELOPMENT COMMAND

Fort Belvoir, Virginia 22060

U.S. ARMY MERADCOM

HYDRAULIC SYSTEM RELIABILITY PROGRAM

HYDRAULIC VALVE SPECIFICATIONS

FINAL REPORT

PROJECT PERSONNEL

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PREFACE

The purpose of the Oklahoma State University/U.S. Army Mobility Equipment Research and Development Command Program is to provide the military with tools for the scientific appraisal of fluid power systems. This report presents the details of project activities concerning hydraulic valve specifications. Tests have been conducted based on Army specification MIL-V-52688 and the test results are discussed. OSU proposed revision of MIL-V-52688 is presented including additional test procedures that are considered necessary for Army valve specification.

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CHAPTER I

SCOPE, PURPOSE AND PLAN OF ATTACK

The hydraulic valve is one of the most important components in the fluid power system. More specifically, the directional control valve is indispensible in construction-type hydraulic mobile equipment. Due to recent technological advancements in the hydraulics industry, many new kinds of directional control valves have been made available. Consequently, adequate specifications to define valve performance requirements are needed.

The scope of this study was to conduct an engineering evaluation of the performance requirements specified in the military specification: Valves, Hydraulic, Directional Control, Spool Type, Manually Operated—MIL-V-52688. It was also an important part of the scope to revise or formalize the performance requirements to make each specification acceptable to industry. In addition to the performance requirements specified in MIL-V-52688, the following documents were to be considered.

Pressure/Flow	Low Temperature
External Control	Performance Modeling
Metering	Structural Integrity
Internal Leakage	Corrosion Resistance
Pressure Response	Contaminant Sensitivity
Durability	Noise Generation Potential

The plan of attack established to accomplish this study was to:

1. Review the performance requirements specified in MIL-V-52688.
2. Evaluate the performance requirements specified in MIL-V-52688 by using the test data.
3. Develop the test procedures necessary to evaluate the performance of directional control valves.
4. Submit the test procedures to the SAE Subcommittee IV to insure their acceptance by industry.

CHAPTER II

CRITIQUE OF CURRENT DIRECTIONAL
CONTROL VALVE SPECIFICATIONS OF MIL-V-52688

TEST PROCEDURES IN MIL-V-52688

1. Spool Operating Force. With a hydraulic pressure 200 psi at the pressure port and cylinder ports blocked, measure the axial force required to move the valve spool from the center position to the full open position. The spool displacement force(s) shall be recorded at rated full flow with oil temperatures of minus 25°F (plus or minus 5°F) and plus 150°F (plus or minus 5°F). An axial force greater than 0.6 pounds for each gpm of rated flow shall constitute failure of this test.

2. Pressure Drop. With an oil temperature of 150°F (plus or minus 5°F) and at rated flow, cycle the valve and record the pressure drop across the valve. A pressure drop greater than that specified in Table 2-1 shall constitute failure of this test.

TABLE 2-1. Pressure Drop (in psi).

Flow gpm	Neutral Position				Operating Position			
	Number of Spools				Number of Spools			
	1	2	3	4	1	2	3	4
10	15	20	25	30	30	40	50	60
20	25	35	45	55	50	70	90	110
30	35	50	65	80	70	100	130	160
40	40	60	80	100	80	120	160	200
50	45	65	85	105	90	130	170	210
60	50	70	90	110	100	140	180	220
70	55	80	105	130	110	160	210	260
80	60	90	120	150	120	180	240	300
90	65	100	135	170	130	200	270	340
100	70	110	150	190	140	220	300	380

3. Internal Leakage. With the valve in the center position, pressurize each actuator port to 2000 psi at an oil temperature of 150°F (plus or minus 5°F). Record the internal leakage of each spool. An internal leakage greater than 2 cc per minute for each gpm of rated flow of the valve shall constitute failure of this test.

4. Proof Pressure. With the spool in the center position and return port open, pressurize the cylinder ports and pressure ports simultaneously to 3000 psi at a rate of not less than 25,000 psi per second and hold for not less than 1 minute. Evidence of permanent deformation, damage or external leakage shall constitute failure of this test.

5. Low Temperature. The test valve shall be installed in a climatic changer with an oil conforming to MIL-L-10295. The valve shall be lowered to minus 50°F (plus or minus 5°F) and maintained at this temperature for 12 hours. At the conclusion of the 12-hour period, the valve shall be cycled with rated flow and maximum operating pressure for a total of 50 cycles, using oil having viscosity of 14,000 SUS. Binding of the spool during 50 cycles shall constitute failure of this test.

6. Endurance Test. Operate the valve through all of its functions at a rate of 20 cycles per minute at rated full flow and at maximum operating pressure for a total of 750,000 cycles. Oil temperature shall be 220°F (plus or minus 5°F) during the first 60,000 cycles and 150°F (plus or minus 5°F) during the balance of the test run. Pressure cycle for the last 200,000 cycles shall be as follows: 5 seconds at

minimum pressure and 5 seconds with pressure limited to not less than 133 percent of the rated pressure. Upon completion of 750,000 cycles, repeat the leakage and pressure drop tests specified herein. Malfunction prior to completion of 750,000 cycles, failure to meet the criteria set forth in the above specified tests, or evidence of external leakage, binding of the spool, scuffing, galling, or other defects shall constitute failure of this test.

7. Salt Spray Test. The valve spool shall be subjected to a 5 percent NaCl solution, salt spray test in accordance with ASTM B117 for a period of not less than 30 hours. Examine the valve spool sealing surfaces for corrosion at conclusion of the test. Staining or corrosion that cannot be removed by rubbing with an oily rag or evidence of pitting shall constitute failure of this test.

8. Abrasive Test. The valve shall be subjected to Procedure I, sand and dust tests, as specified in MIL-E-5272, except that the sand and dust used in the test shall conform to SAE J726, AC Test Dust, Fine. The valve shall be cycled for 10 hours at a rate of 20 cycles per minute. At the conclusion of the test, examine the valve for wear and damage. Upon completion of this test, repeat the pressure drop and internal leakage tests specified herein. Inability of the valve to meet the criteria set forth in the above specified tests shall constitute failure of this test.

TEST EXPERIENCES

Three identical open-center, manually-operated, spool-type directional control valves—P-1, P-2, and P-3—whose rated flow is 20 gpm

and the rated pressure is 2000 psi and four identical, open-center, pilot-operated, spool-type, directional control valves—H-1, H-2, H-3 and H-4—whose rated flow is 20 gpm and the rated pressure is 2000 psi were tested by the test procedures specified in MIL-V-52688 at the Fluid Power Research Center.

1. Spool Operating Force. Valves P-1, P-2, and P-3 were tested for spool operating force. The cylinder ports were plugged and the valve was operated at 20 gpm and 2000 psi at a temperature of 150°F (plus or minus 5°F). The force required to actuate the spool was recorded as a function of displacement on an X-Y recorder. The maximum force was then read from the resulting plot. The temperature was lowered to -25°F (plus or minus 5°F) and the valve was retested. The maximum forces to operate spools are as follows:

TABLE 2-2. Maximum Spool Operating Force for Valves P-1, P-2, and P-3.

Spool Number	Spool In	Spool Out
P-1	74 lbs	53.4 lbs
P-2	65 lbs	69.0 lbs
P-3	68 lbs	109.2 lbs

Oil Temperature = 150°F

Spool Number	Spool In	Spool Out
P-1	91.0 lbs	84.6 lbs
P-2	75.0 lbs	77.4 lbs
P-3	98.0 lbs	160.0 lbs

Oil Temperature = -25°F

As a result of these tests, none of the three valve spools met the operating force requirement of 12 lbs or less at a temperature of 150°F or at a temperature of -25°F.

2. <u>Pressure Drop</u>. Valves P-1, P-2, P-3, H-1, H-2, H-3, and H-4 were tested for pressure drop. The cylinder ports were connected together and the valve was operated in each position at 20 gpm, 2000 psi and 150°F (plus or minus 5°F). The pressure drop across the total valve was then measured and recorded. The tabulated results are shown in Table 2-3.

TABLE 2-3. Results of Pressure Drop Test.

Spool Number	Pressure Drop Port A	Pressure Drop Port B
P-1	200.0 psi	162.5 psi
P-2	182.5 psi	182.5 psi
P-3	197.0 psi	216.0 psi
H-1	86.0 psi	90.0 psi
H-2	45.0 psi	115.0 psi
H-3	45.0 psi	114.0 psi
H-4	105.0 psi	99.0 psi

Valves P-1, P-2, and P-3 failed to meet the pressure drop requirement of less than 130 psi, and valves H-2 and H-3 failed to meet the requirement of less than 110 psi. Valves H-1 and H-4 met the requirement of less than 110 psi pressure drop.

3. <u>Internal Leakage</u>. Valves P-1, P-2, P-3, H-1, H-2, H-3 and H-4 were tested for internal leakage. Each valve was fitted with a small valve at the cylinder port while all other cylinder ports and the return port were plugged. Pressure was then applied to the valve at the inlet at 2000 psi and 150°F (plus or minus 5°F) with the valve spool centered. The small valve at the cylinder port was opened and the leakage measured with a graduated cylinder and stop watch. The test results are presented in Table 2-4.

TABLE 2-4. Results of Internal Leakage Tests.

Spool Number	Internal Leakage Port A	Internal Leakage Port B
P-1	5.67 mℓ/min	8.0 mℓ/min
P-2	25.0 mℓ/min	25.33 mℓ/min
P-3	>40.0 mℓ/min	>40.0 mℓ/min
H-1	10.5 mℓ/min	15.0 mℓ/min
H-2	17.5 mℓ/min	15.5 mℓ/min
H-3	17.0 mℓ/min	14.5 mℓ/min
H-4	13.0 mℓ/min	28.5 mℓ/min

Valves P-1, P-2, H-1, H-2, H-3, and H-4 met the requirement of less than 40 mℓ/min leakage, whereas valve P-3 did not.

4. Proof Pressure. Valves P-1, P-2, P-3, H-1, H-2, H-3 and H-4 were proof tested. With the valve in the center position, the valve was pressurized with fluid at 3000 psi at a rate of at least 30,000 psi per second. The pressure rise rate was recorded with a light oscillograph. The valve was held at pressure in excess of one minute, at which time the valve was checked for any external leakage or observable damage.

All the valves met the requirement of 25,000 psi per second and sustained pressure of 3000 psi for one minute without apparent damage.

5. Low Temperature. Valves P-1, P-2, P-3, H-1, H-2, H-3 and H-4 were tested at low temperature. Dry ice was placed in an insulated chamber and the surrounding air allowed to cool to -50^{o}F (plus or minus 5^{o}F). MIL-L-46167-type oil was then circulated throughout the valve's chambers. At this time, the valve was placed into the cold environment. At the end of twelve hours in the cold environment, the valve was cycled through its positions with rated flow and maximum

operating pressure for a total of 50 cycles. It was then inspected for any binding or galling of any of the spools.

Each spool met the requirement of the low temperature environment without any binding of the spool.

6. <u>Endurance Test</u>. Valves P-1, P-2, P-3, H-1, H-2, H-3, and H-4 were tested. The valve was mounted in the test fixture. All cylinder ports were plugged and all spools were connected to a single actuating arm. The valve was then operated for a total of 750,000 cycles at 20 gpm with the following restrictions:

a. For the first 60,000 cycles, the oil temperature was 220^{O}F (plus or minus 5^{O}F) with a cycling rate of 20 cycles/in. at a pressure of 2000 psi.

b. During the next 490,000 cycles, the oil temperature was 150^{O}F (plus or minus 5^{O}F) and the cycling rate was 20 cycles per minute at a pressure of 2000 psi.

c. The last 200,000 cycles were run with an oil temperature of 150^{O}F (plus or minus 5^{O}F), a cycle rate of 6 cycles per minute and a pressure of 2600 psi.

At the end of the test, the pressure drop test and internal leakage tests were performed. The results are as follows:

TABLE 2-5. Results of Pressure Drop Test Performed After the 750,000 Cycle Endurance Test.

Spool Number	Pressure Drop Port A	Pressure Drop Port B
P-1	632.5 psi	632.5 psi
P-2	625 psi	625 psi
P-3	575 psi	575 psi
H-1	99 psi	96 psi
H-2	hit relief valve setting	125 psi
H-3	67 psi	115 psi
H-4	106 psi	98 psi

TABLE 2-6. Results of Leakage Tests Performed After the 750,000 Cycle Endurance Test.

Spool Number	Leakage Port A	Leakage Port B
P-1	41.0 mℓ/min	23.0 mℓ/min
P-2	72.0 mℓ/min	57.0 mℓ/min
P-3	>40.0 mℓ/min	>40.0 mℓ/min
H-1	11.3 mℓ/min	12.0 mℓ/min
H-2	26.0 mℓ/min	>40.0 mℓ/min
H-3	18.5 mℓ/min	13.0 mℓ/min
H-4	10.5 mℓ/min	8.5 mℓ/min

All valves except valve P-3 and valve H-2, port B, failed the leakage test. Valves P-1, P-2, P-3, H-2 and H-3 failed the pressure drop test. There were not signs of external damage.

7. Salt Spray Test. Valves P-1, P-2, and P-3 underwent the salt spray test. Each valve was disconnected from the hydraulic stand and cleaned. It was then placed in the salt spray chamber and subjected to a salt mist in accordance with ASTM B117 for a period of 30 hours. The valve was removed and checked for any staining or pitting that could not be removed with aid of an oily rag. The valves passed all requirements of the salt spray performance criterion.

8. Abrasive Test. Valves P-1, P-2, and P-3 underwent the abrasive test. Each valve was cleaned and placed into the dust chamber. The chamber was then sealed the the valve subjected to an abrasive environment as specified in MIL-E-5272C(ASG) Procedure I, except the dust used was SAE J726 AC Fine Test Dust and the valve was cycled for 10 hours at a rate of 20 cycles/min. The valve was removed at the completion of the test and then tested for pressure drop and internal leakage. The results were as follows:

TABLE 2-7. Results of Pressure Drop Test Performed After the Ten-Hour Abrasive Test.

Spool Number	Pressure Drop Port A	Pressure Drop Port B
P-1	420 psi	340 psi
P-2	380 psi	330 psi
P-3	300 psi	290 psi

TABLE 2-8. Results of Leakage Test Performed After the Ten-Hour Abrasive Test.

Spool Number	Leakage Port A	Leakage Port B
P-1	20.7 mℓ/min	21.7 mℓ/min
P-2	46.3 mℓ/min	59.3 mℓ/min
P-3	>40 mℓ/min	>40 mℓ/min

All valves failed the pressure drop test. Valve P-1 passed the leakage test while valves P-2 and P-3 failed the test. There was external leakage at the rear of spool of valve P-1.

RECOMMENDATIONS ON MIL-V-52688 PROCEDURES BASED ON THE TEST EXPERIENCES

1. Spool Operating Force. The test for the actuating force was not clear in the purpose as the test valve was closed centered and as such gave no reason for the valve force to change with the cylinder ports plugged. Also, the spools had detents and were spring-centered

which affects the actuating force to a large extent. No provisions were made to distinguish between these differences.

2. Pressure Drop. The energy cost for directing fluid is too high. For example, a 100-gpm valve with one spool dissipates 4 hp of energy at the neutral position and 8 hp at the operating position.

3. Internal Leakage. The definition on how the leakage tests were to be performed were vague, since spool design again affects this drastically.

4. Proof Pressure. The strain damage during 25,000 psi/sec rise is trivial compared with the damage for at least 60 sec at proof pressure. The proof pressure of 3,000 psi is considered too low.

5. Low Temperature. For binding sensitivity, arctic oil should be used in the test. MIL-2104 at -25^{O}F cannot be pumped becuase its pour point is -15^{O}F.

6. Endurance Test. The endurance test was completely obscure in its purpose and description of execution. Thistest definitely is ambiguous and many actions taken during the test seem to lack purpose.

7. Salt Spray Test. This test was achieved satisfactorily with a specially fabricated chamber.

8. Abrasive Test. The abrasive test does not seem to take into account the fact that oil will be present under normal operating conditions, and therefore should be included with the test.

CHAPTER III

DEVELOPMENT OF CONTAMINANT LOCK TEST

Contaminant lock is one of the most serious problems which can occur using spool-type directional control valves since it is usually accompanied by a catastrophic-type failure. The possibility for lock to occur is due to the fact that all contaminants cannot be removed from a system. Also, the spool clearance cannot be made large enough to tolerate all the contaminants involved.

Since the military specification (MIL-V-52688) does not include the contaminant lock sensitivity test procedure, it was necessary to develop one with the appropriate interpretation techniques.

TEST PROCEDURE DISCUSSSION AND PRESENTATION

1. Test Facilities

In order to evaluate the contaminant lock characteristics of spool valves, a special test facility was constructed. Fig. 3-1 is a schematic of the test circuit. The directional control valve was installed in the test circuit, and suitable measurement equipment was attached to the valve as shown in Fig. 3-2. The spool travel distance and the spool operating force were measured by displacement transducers and force transducers, respectively. Both measurements were recorded by an X-Y recorder. Since the movement of the spool for measurement purposes needed to be kept as small as possible to prevent losing the contaminant clogged in the spool clearance, a spool control device was used to allow less than 1/100-inch spool travel.

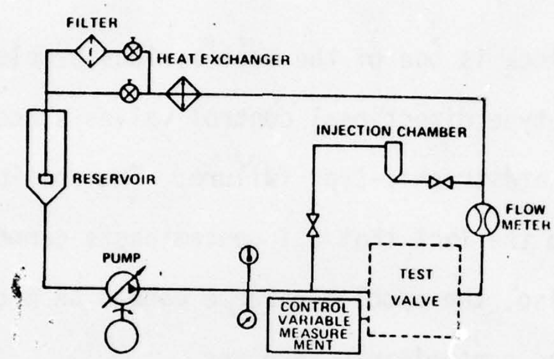


Fig. 3-1. Contaminant Lock Test Circuit.

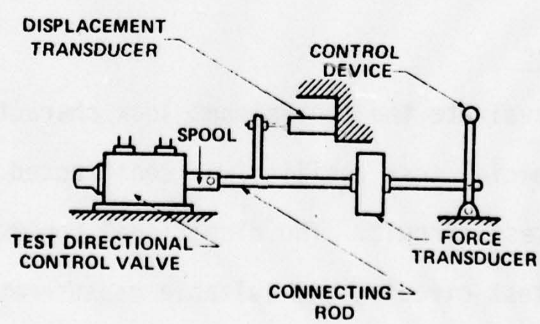


Fig. 3-2. Measurement Equipment.

2. Test Conditions

To achieve accurate measurements of the contaminant silting force on the spool, the valve's centering spring was always removed before starting the test. The test circuit was operated at constant pressure—100 psi (69 bar), constant flow rate—15 gpm (57 ℓpm), and constant temperature—150^{o}F (66^{o}C). The spool was moved to the extreme end of its operating position. All measurements were achieved at this spool position.

The spool friction force was measured prior to injecting contaminant by a finite movement of the spool with clean fluid. After removing the filter from the flow stream, the test contaminant was introduced into the test circuit. The contaminant silting forces on the spool, in addition to the spool friction force, were measured as a function of stationarity time.

The stationarity time is defined as the time period from initial contaminant injection to the time that the spool operation or movement is attempted.

The contaminant silting force is defined as the additional force needed to shift the valve spool after exposure to a given contaminant environment.

3. Test Contaminants

To derive the most discriminative test results, two test contaminants were considered: lower cut AC Fine Test Dust and upper cut AC Fine Test Dust. Table 3-1 shows the particle size distribution of lower cut dust while Table 3-2 shows the upper cut dust. The lower

THE LOWER CUT DUST

INTERVAL µm	µm 0-10	0-20	0-30	0-40	0-50	0-60	0-70
0/5	74.2%	68.7%	67.9%	67.8%	67.7%	67.7%	67.7%
5/10	25.8	23.9	23.6	23.6	23.5	23.5	23.5
10/20		7.4	7.4	7.4	7.4	7.4	7.4
20/30			1.1	1.0	1.0	1.0	1.0
30/40				0.2	0.3	0.26	0.26
40/50					0.1	0.11	0.10
50/60						0.03	0.03
60/70							0.01

TABLE 3-1. Particle Distribution of the Lower Cut Dust.

THE UPPER CUT DUST

INTERVAL µm	µm 5-UP	10-UP	20-UP	30-UP	40-UP	50-UP	60-UP
5/10	72.1%						
10/20	22.9	82.3%					
20/30	3.5	12.5	70.6%				
30/40	0.9	3.2	18.2	61.8%			
40/50	0.3	1.1	6.2	21.0	55.0%		
50/60	0.1	0.45	2.5	8.6	22.4	49.8%	
60/70	0.06	0.20	1.2	3.9	10.3	22.8	45.5%
70/80	0.03	0.10	0.58	2.0	5.2	11.5	22.9
90/UP	0.11	0.15	0.72	2.7	7.1	15.9	31.6

TABLE 3-2. Particle Distribution of the Upper Cut Dust.

cut dust has a high proportion of 0-5 μm interval size particles as shown in Table 3-1. Note that the number of "large" particles is small, even for large classification sizes. For example, 0-50 μm contaminant includes 67.7% of 0-5 μm size contaminant and only 0.1% of 40-50 μm size contaminant. On the other hand, "upper cut dust," as shown in Table 3-2, has a large number of the first interval size particles. That is, 40 μm-up contaminant includes 55% of just 40-50 μm size contaminant.

Tests were conducted with both "lower cut dust" and "upper cut dust" at the contaminant concentration of 100 mg/ℓ. Figures 3-3 and 3-4 show the test results of the lower cut dust and of the upper cut dust, respectively. It is apparent from the figures that the upper cut dust shows the most discriminatory results. Therefore, upper cut dust was selected as the test contaminant to be used for the contaminant lock test.

4. Test Procedure

Figure 3-5 shows the sequence of operations for the contaminant lock test. Table 3-3 illustrates all steps necessary to satisfactorily complete the test.

Before injecting contaminant into the test system, all operating parameters must be stabilized at the specified levels. The spool friction force is then measured in the initially "clean" environment. The quantity of contaminant needed to be injected for a gravimetric level of 50 mg/ℓ in the system must be determined. The system control filters are isolated from the circuit and the required amount of 0-5 μm

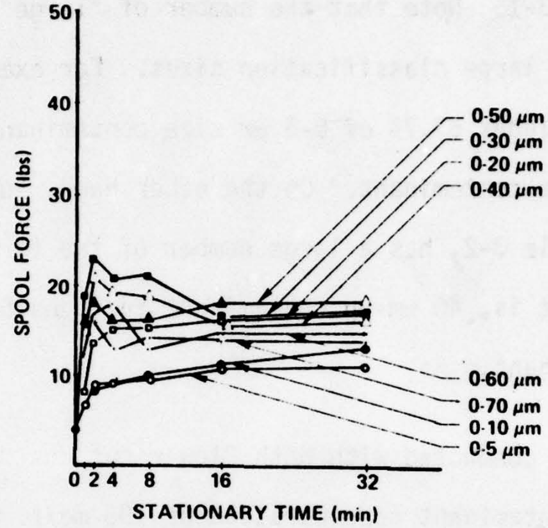


Fig. 3-3. Spool Force Measurements with Lower Cut Dust.

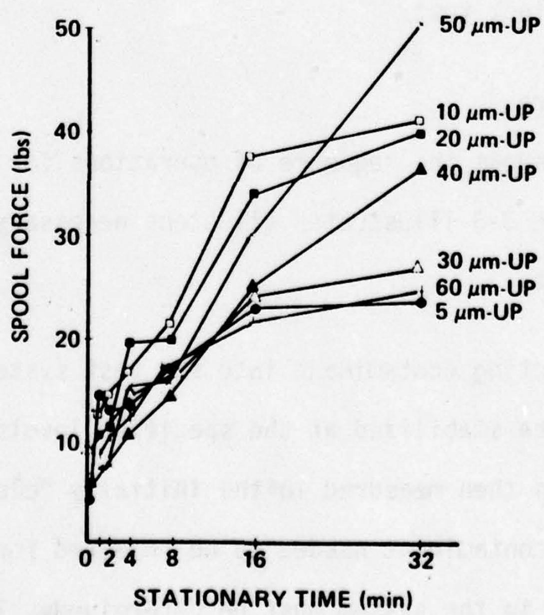


Fig. 3-4. Spool Force Measurements with Upper Cut Dust.

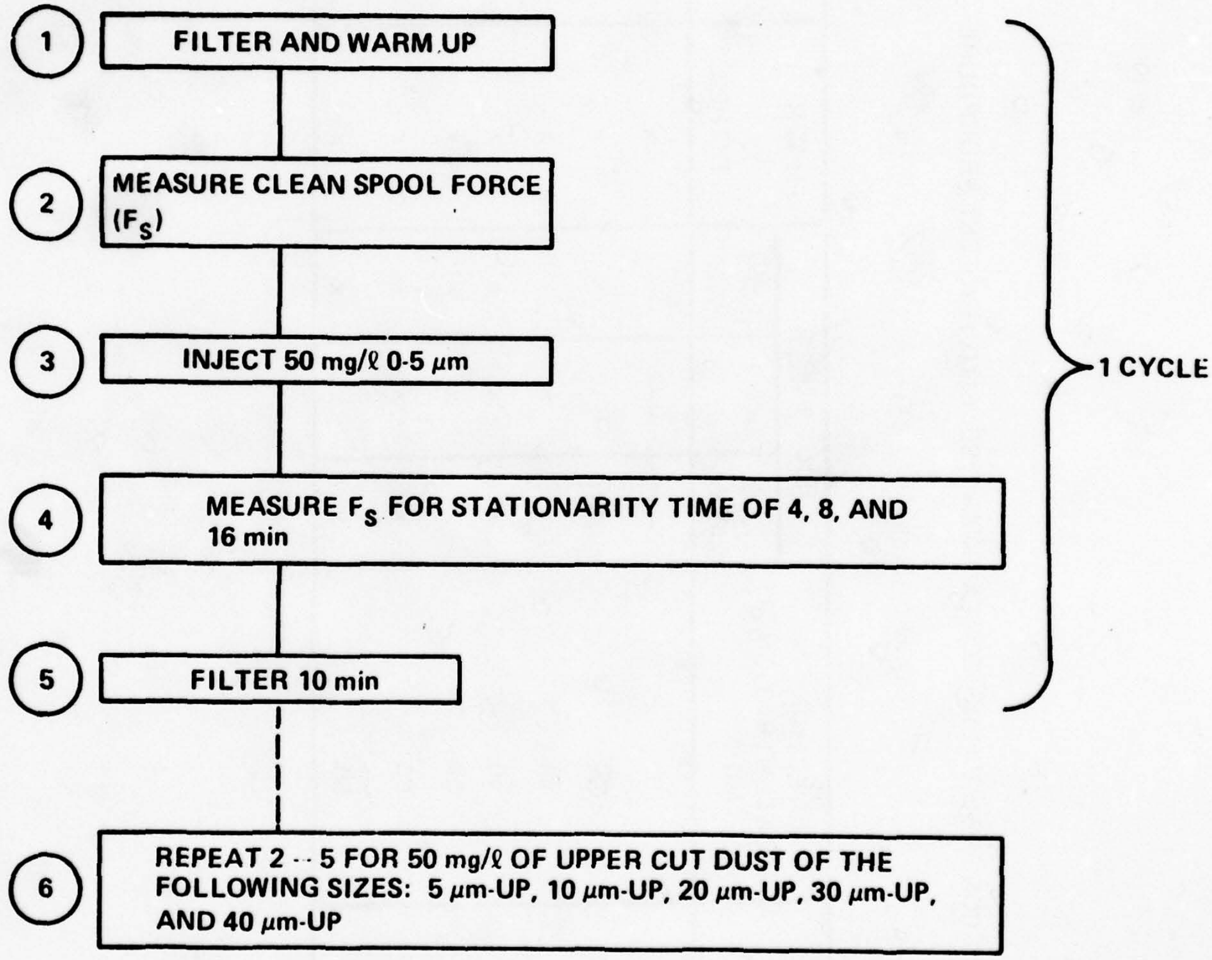


Fig. 3-5. Contaminant Lock Test Procedure.

DIRECTIONAL CONTROL VALVE CONTAMINANT SENSITIVITY TEST PROCEDURE

OSU-V-17

STEP	INJECTION SIZE μM	INJECTION CONCENTRATION MG/ℓ	SILT FORCE 4 MIN.	SILT FORCE 8 MIN.	SILT FORCE 16 MIN.	FILTER DITHER	SPOOL FORCE
1	None		—	—	—	x	x
2	0-5	50	x	x	x	x	x
3	5-Up	50	x	x	x	x	x
4	10-Up	50	x	x	x	x	x
5	20-Up	50	x	x	x	x	x
6	30-Up	50	x	x	x	x	x
7	40-Up	50	x	x	x	x	x

TABLE 3-3.

contaminant is injected into the working fluid. The operating parameters are maintained constant as the contaminant is circulated through the system. The spool is kept stationary except for minute movements when measuring the degree of contaminant lock. Contaminant silting forces are measured at 4, 8 and 16 minutes from the time the contaminant is first injected. These periods are termed "stationarity times." Following these three measurements, the system control filters are connected back into the test circuit and the fluid is filtered for 10 minutes (or the necessary period required to obtain a "clean level"). This sequence of steps is repeated until the particle size injection of 40 μm-up has been completed.

5. Discrimination and Repeatability of the Test

The contaminant lock test procedure has been conducted numerous times on various manufacturers' valves to compile data needed to verify the integrity of the recommended test procedure. Fig. 3-6 shows the graphical results of two such tests obtained on two valves identified as OSU Valve No. 104-1 and OSU Valve No. 104-2. The obvious dissimilarity in the contaminant lock characterisitics of the two valves illustrates the degree of discrimination which the test provides. This ability of the test to discriminate is futher exemplified when it is realized that the two valves use the same valve housing. The only difference between them is the amount of spool clearance.

Excellent repeatability of the test procedure is apparent in Fig. 3-7 which shows the results of two tests where the procedure is repeated at the same operating conditions on OSU Valve No. 103.

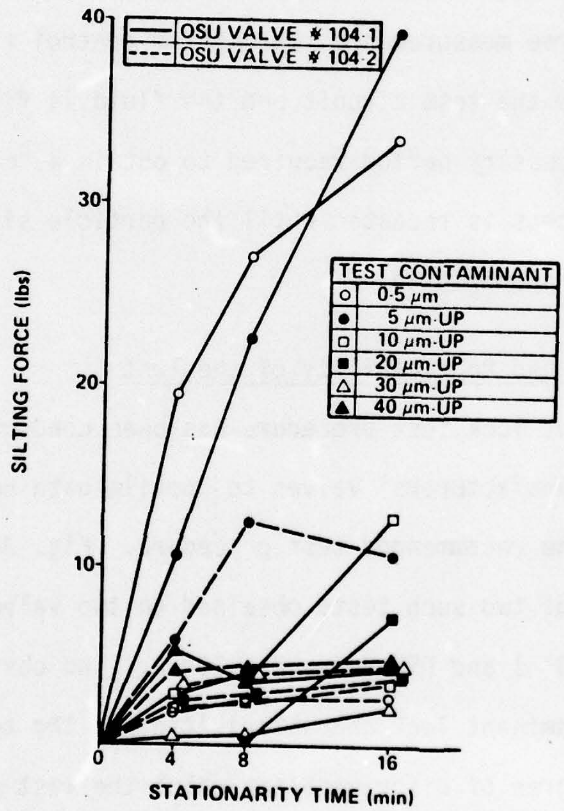


Fig. 3-6. Discrimination Capability of Contaminant Lock Test.

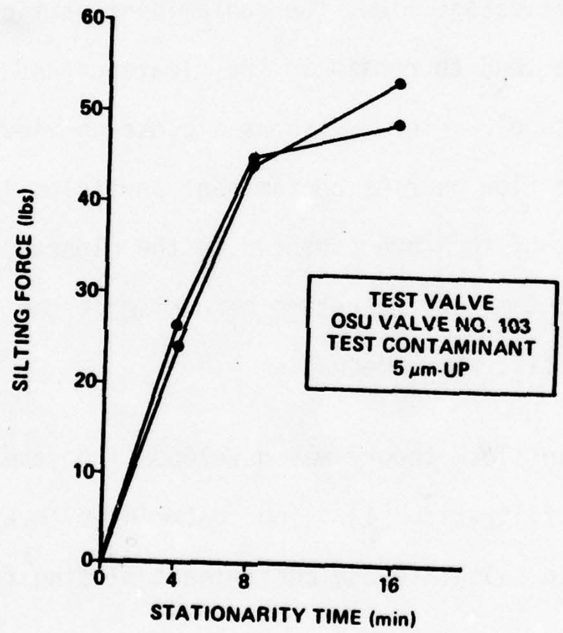


Fig. 3-7. Repeatability Test Data.

CONTAMINANT LOCK THEORY

The contaminant lock theory was developed for the interpretation technology of the contaminant lock test results. Fig. 3-8 shows a common structure of spool-type directional-control valves. Since the clearance between bore and spool is usually minimized to reduce the amount of internal leakage flow, the contaminant particles passing into the clearance tend to remain in the clearance and cause contaminant lock of the spool. Fig. 3-9 shows a close-up view of the clearance. The leakage flow carries contaminant particles into the clearance and some of them are captured in the clearance as shown in Fig. 3-9. This contaminant retention mechanism is the same as the constant-pressure filtration mechanism.

The contaminant lock theory was developed from the concepts of constant-pressure filtration [1]. The contaminant lock theory gives Eq. (3-1) which can calculate the contaminant silting force (derived in Ref. [1]).

$$F = a\pi D \ell h_o \left(1 - \frac{1}{\sqrt{\frac{\epsilon V_p \Delta P h_o^2}{6(1-\phi)\mu \ell^2}t + 1}}\right) \qquad (3\text{-}1)$$

where: F = silting force

a = silting force constant

D = spool diameter

ℓ = effective clearance length

h_o = initial effective clearance

ε = contaminant retention efficiency

V_p = contaminant volume per unit liquid volume

ΔP = pressure differential across the clearance

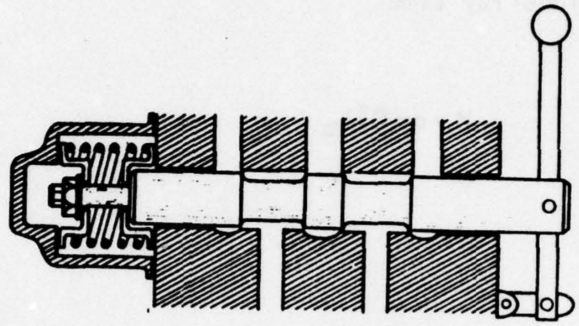


Fig. 3-8. Spool-Type Directional-Control Valve.

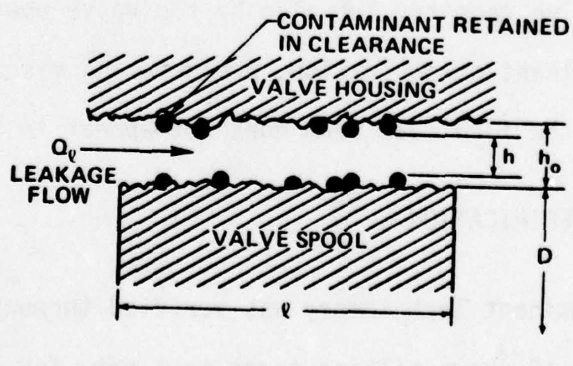


Fig. 3-9. Contaminant Retention in Spool Valve Leakage Path.

ϕ = porosity of compacted particles

μ = absolute viscosity of fluid

t = stationarity time

When we set

$$X_1 = a \pi D \ell h_0 \tag{3-2}$$

and

$$Y_1 = \frac{\epsilon V_p \Delta P h_0^2}{6(1-\phi)\mu \ell^2} \tag{3-3}$$

Equation (3-1) is simplified to

$$F = X_1 \left(1 - \frac{1}{\sqrt{Y_1 t + 1}}\right) \tag{3-4}$$

Equation (3-4) shows the silting force F is a function of stationarity time and two parameters X_1 and Y_1. Parameter X_1 is affected by only the valve geometry. On the other hand, parameter Y_1 is affected not only by the valve geometry but also by the valve operating parameters such as contaminant concentration, pressure and viscosity. It should be noted that the flow rate term does not appear in the equation.

EXPERIMENTAL VERIFICATION

The contaminant lock theory was verified through many experimental tests. Fig. 3-10 shows silting force test data for a commercial spool valve (OSU Valve No. 102) and the theoretical curve for the valve as expressed by Eq. (3-1). A good correlation between the test data and the theoretical curve is observed.

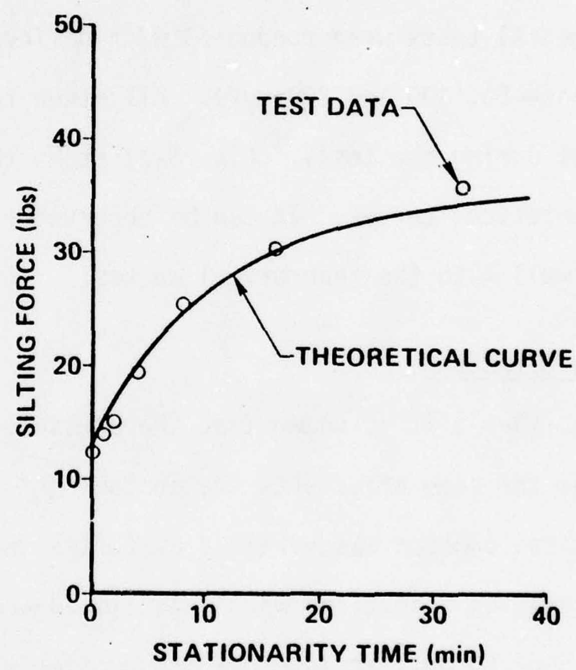


Fig. 3-10. Theoretical and Experimental Silting Force of Spool Valve No. 102 with Contaminant 30 μm-up at 100 mg/ℓ.

1. Contaminant Concentration Effects

From Eq. (3-3), altering the contaminant concentration results in a change in the value of Y_1. Doubling the contaminant concentration doubles the value of Y_1 in Eq. (3-4).

Experimental tests were conducted with different contaminant concentrations—50, 100 and 200 mg/ℓ. All other test parameters were kept constant during the tests. Fig. 3-11 shows the results compared with the theoretical curves. It can be observed that the test data agree quite well with the theoretical curves.

2. Pressure Effects

From Eq. (3-3), it is known that the pressure differential across the spool has the same effects as the contaminant concentration has. This theoretical concept was verified by further experiments. Two different operating conditions were established—one with a contaminant concentration of 50 mg/ℓ at 1000 psi and another with 100 mg/ℓ at 500 psi. The second contaminant concentration was doubled, while the operating pressure was reduced to half of the first condition. All other operating parameters were held constant. These two different operating conditions produce the same Y_1 value in Eq. (3-4) which means that the silting forces should be identical.

Figure 3-12 shows the experimental results. The silting forces for the two different conditions were extremely close for the stationarity time of 0, 4, and 8 minutes. But the silting forces for the low concentration per high pressure condition were lower than predicted at the longer stationarity times. It is suspected that undesirable

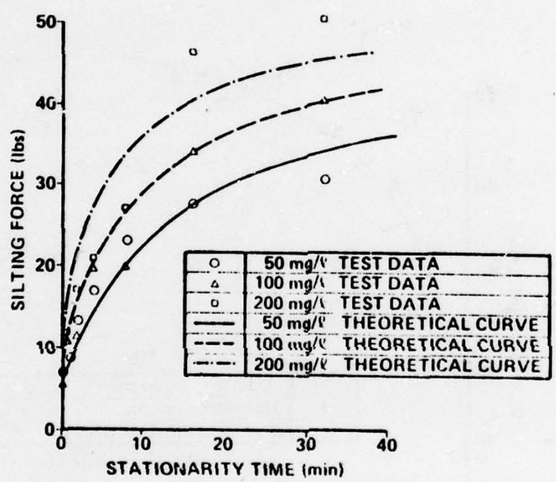


Fig. 3-11. Contaminant Concentration Effect on the Silting Force of Spool Valve No. 101 with Contaminant 20 μm-up.

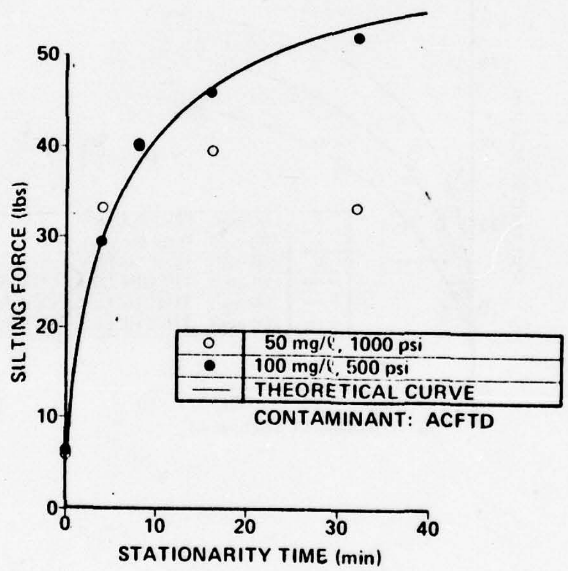


Fig. 3-12. Silting Forces of Spool Valve No. 102 in Two Different Conditions with the Same Y_1.

operating factors existed, such as excessive movement of the spool or vibration which allowed contaminant to escape from the spool clearance which, in turn, resulted in reduction of the silting force. In practice, the silting force curve can be expected to follow very closely the theoretical silting force curve.

3. Flow Effects

The system flow rate does not have any effect on the contaminant silting force because it is the leakage flow through the spool clearance that transports the contaminants into the clearance space and the leakage flow is governed only by the pressure differential, fluid viscosity and valve geometry. To verify this, two different system flow rates were established—15 gpm and 7.5 gpm, with all other operating parameters held constant. Fig. 3-13 shows the test results.

Theoretically, these two operating conditions are expected to result in the same silting forces. However, the silting forces at the 7.5 gpm condition were always observed to be less than that at 15 gpm. It is apparent that particle impingement and inertial effects have some influence which are yet to be included. Particle impingement into the spool clearance or the inertial effect of the fluid on the particles can obviously force contaminant to lodge in the clearance opening and it might be expected that the severity of the particle lodgement at 15 gpm is greater than at 7.5 gpm. Therefore, it is not unreasonable to observe silting forces at 7.5 gpm.

In reality, the difference between the silting forces at the two operating conditions reflected in Fig. 3-13 is negligible when consider-

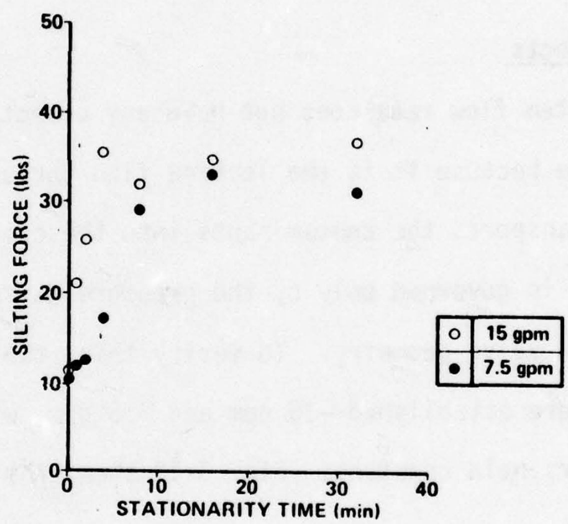


Fig. 3-13. Silting Forces of Spool Valve No. 103 Under Two Different Flow Rates with Contaminant 40 μm-up at 100 mg/ℓ.

ing the degree of experimental error. Hence, Eq. (3-1), which excludes any flow rate term, is a valid means of predicting the contaminant lock which can occur under field operating conditions.

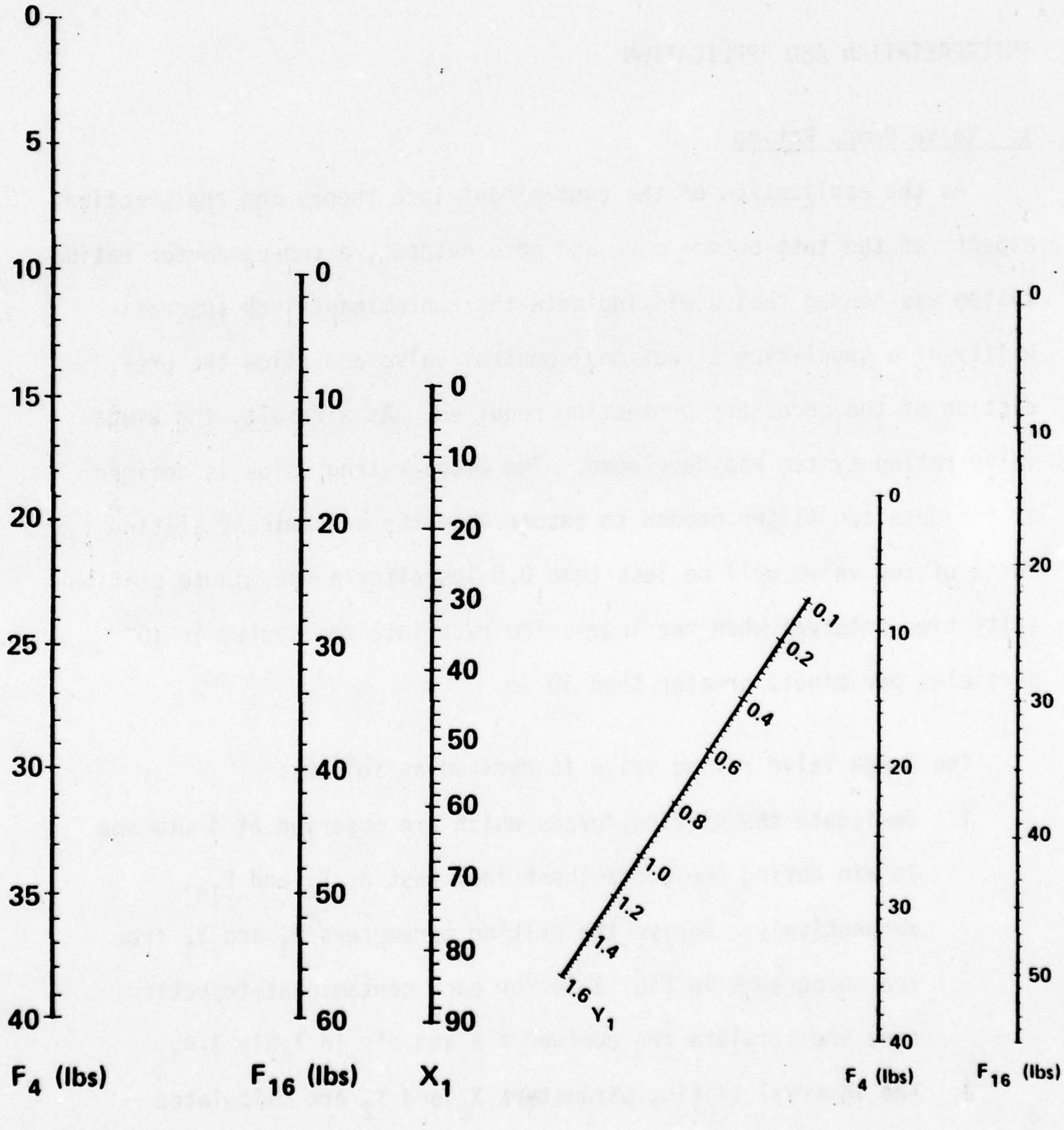
INTERPRETATION AND APPLICATION

1. Valve Omega Rating

As the application of the contaminant lock theory and the practical aspects of the test become more and more evident, a one-parameter rating system was needed that could indicate the contaminant lock susceptibility of a spool-type directional-control valve and allow the prediction of the necessary protection required. As a result, the Omega valve rating system was developed. The Omega rating value is defined as the Beta ten filter needed to ensure that the contaminant silting force of the valve will be less than 0.5 lbs after a one-minute stationarity time interval when the ingression rate into the system is 10^8 particles per minute greater than 10 μm.

The Omega Valve rating value is derived as follows:

1. Designate the silting forces which are observed at 4 min and 16 min during the contaminant lock test as F_4 and F_{16}, respectively. Derive the silting parameters X_1 and Y_1 from the nomographs in Fig. 3-14 for each contaminant injection size and tabulate the derived X's and Y's in Table 3-4.
2. The interval silting parameters X_2 and Y_2 are calculated using Table 3-5 and 3-6.
3. The reference silting parameters X_r and Y_r which give the silting force in the contaminant environment controlled by



(a) (b)

Fig. 3-14. Silting Parameter Nomographs.

INJECTION SIZE (µm)	SINGLE CUT SILTING PARAMETERS			
	X_1		Y_1	
0-5	$X1_{0\text{-}5}$		$Y1_{0\text{-}5}$	
5-UP	$X1_{5\text{-}UP}$		$Y1_{5\text{-}UP}$	
10-UP	$X1_{10\text{-}UP}$		$Y1_{10\text{-}UP}$	
20-UP	$X1_{20\text{-}UP}$		$Y1_{20\text{-}UP}$	
30-UP	$X1_{30\text{-}UP}$		$Y1_{30\text{-}UP}$	
40-UP	$X1_{40\text{-}UP}$		$Y1_{40\text{-}UP}$	

TABLE 3-4. Silting Parameters.

INJECTION SIZE (μm)	X_1	MULTI-FACTOR	PRODUCT	GROSS DIFFERENCE	DIVISION FACTOR	X_2	INTERVAL SIZE (μm)
5-UP							0-5
					0.33		5-10
10-UP		X 0.67					
					0.45		10-20
20-UP		X 0.55					
					0.35		20-30
30-UP		X 0.65					
					0.33		30-40
40-UP		X 0.67					40-UP

$0\text{-}5\,X_2 = 0\text{-}5\,X_1$ ALSO $40\text{-UP}X_2 = 40\text{-UP}\,X_1$

TABLE 3-5. Interval Silting Parameter X_2.

INJECTION SIZE (μm)	Y_1	MULTI-FACTOR	PRODUCT	GROSS DIF-FERENCE	DIVISION FACTOR	Y_2	INTERVAL SIZE (μm)
5-UP							0-5
					0.33		5-10
10-UP		X 0.67					
					0.45		10-20
20-UP		X 0.55					
					0.35		20-30
30-UP		X 0.65					
					0.33		30-40
40-UP		X 0.67					40-UP

$0\text{-}5Y_2 = 0\text{-}5\,Y_1$ ALSO $40\text{-UP}Y_2 = 40\text{-UP}Y_1$

TABLE 3-6. Interval Silting Parameter Y_2.

a β_{10} = 1.1 filter, are computed in Table 3-7 and 3-8.

4. The Omega valve rating value is derived using the nomograph in Fig. 3-15.

In the process of deriving the Omega valve rating value, the interval silting parameter X_2 indicates the maximum silting force that the specified interval size contaminant would cause at an infinite stationarity time. Hence, X_2 can be considered as a degree of contaminant lock sensitivity at a given interval size contaminant. Fig. 3-16 shows a comparison of the values of X_2 obtained from various spool-type directional-control valves. As can be noted, OSU Valve No. 104-1 shows an exceptionally high sensitivity at the interval size 0-20 μm. Generally speaking, those valves whose sensitivity reaches a maximum over the intervals of larger contaminant sizes can more easily be protected by simple filtration of the fluid system than those valves whose peak occurs at small contaminant sizes. This is due to the inefficiency of filters to capture small particles as compared to their ability to trap large sizes. From this standpoint, OSU Valve No. 101 can be more easily protected from its critical particle sizes than OSU Valve No. 104-1 and OSU Valve No. 103.

2. Filter Requirements to Protect Valve

It is apparent that with a higher quality filter placed in a system, the magnitude of the silting force generated in a given spool-type directional-control valve is decreased. By knowing the Omega valve rating and the required system Beta ten filter for a given system ingression rate, a specified contaminant silting force for any stationarity time can be calculated. Fig. 3-17 is the nomograph to

SIZE	X_2	MULTI-FACTOR	PRODUCT
0-5		X 0.68 =	①
5-10		X 0.24 =	②
10-20		X 0.08 =	③
		① + ② + ③ =	X_r

TABLE 3-7. Reference Silting Parameter X_r.

SIZE	Y_2	MULTI-FACTOR	PRODUCT
0-5		X 0.68 =	①
5-10		X 0.24 =	②
10-20		X 0.08 =	③
		① + ② + ③ =	Y_r

TABLE 3-8. Reference Silting Parameter Y_r.

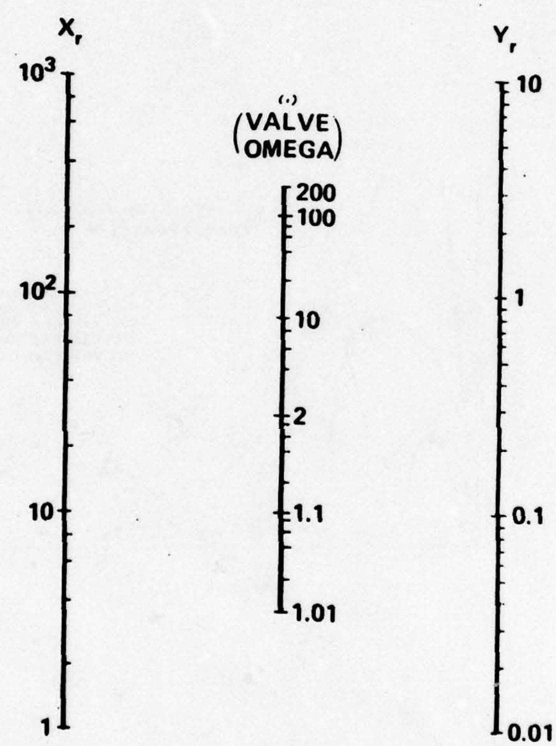


Fig. 3-15. Valve Omega Nomograph.

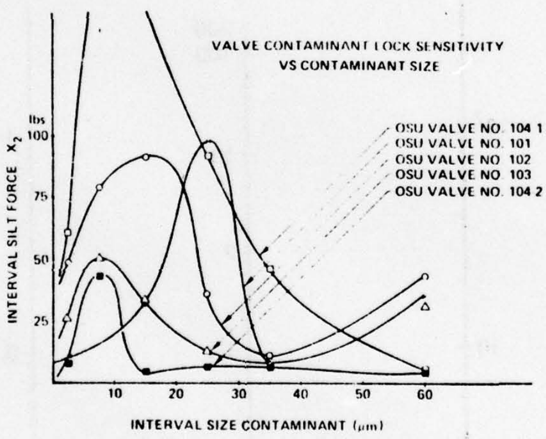


Fig. 3-16. Valve Contaminant Lock Sensitivity.

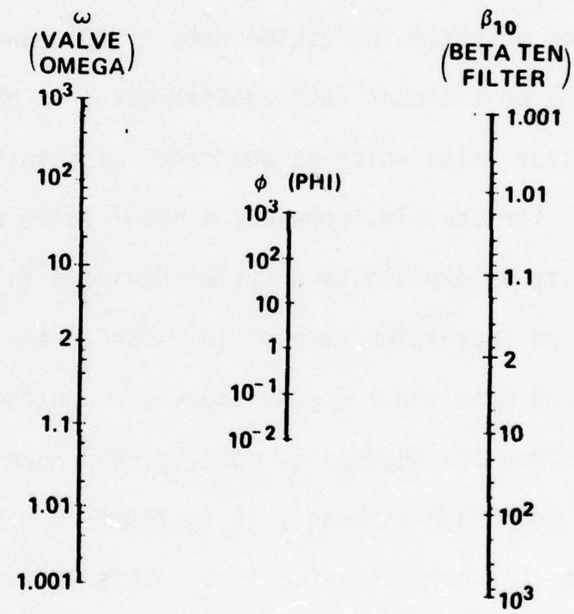


Fig. 3-17. Contaminant Lock Coefficient Nomograph.

derive the contaminant lock coefficient ϕ from the Omega valve rating and the given system filter Beta ten for an ingression rate of 10^8. The contaminant lock coefficient indicates the severity of contaminant lock at a given condition.

Although the nomograph, Fig. 3-17, is designed for an ingression rate of 10^8, for a system ingression rate of 10^7 (one order of magnitude lower), the contaminant lock coefficient ϕ is obtained by using a Beta then filter value which is one order of magnitude higher than actually used. For example, consider a spool valve whose Omega rating is 2, which is to be exposed to a system having a filter with a Beta ten of 1.1 and an ingression rate of 10^7. Referring to Fig. 3-17 for Omega = 2 and Beta ten = 2 (one order of magnitude higher than the actual filter since the ingression rate is one order of magnitude lower than the nomograph assumes), it is found that the value of the contaminant lock coefficient ϕ is 1. This manipulation of Fig. 3-17 is also applicable for ingression rates greater than 10^8 by assuming a Beta ten value less than that actually used.

With the contaminant coefficient ϕ known from the nomograph, Fig. 3-17, the silting force at any stationarity time can be derived by the nomograph in Fig. 3-18. Referring to the previous example where ϕ = 1, the silting force for a stationarity time of 10 min is determined to be 4.5 lbs.

To predict the amount of protection needed for a given valve at specified conditions, the following procedure should be followed. Consider a spool-type directional-control valve whose Omega rating

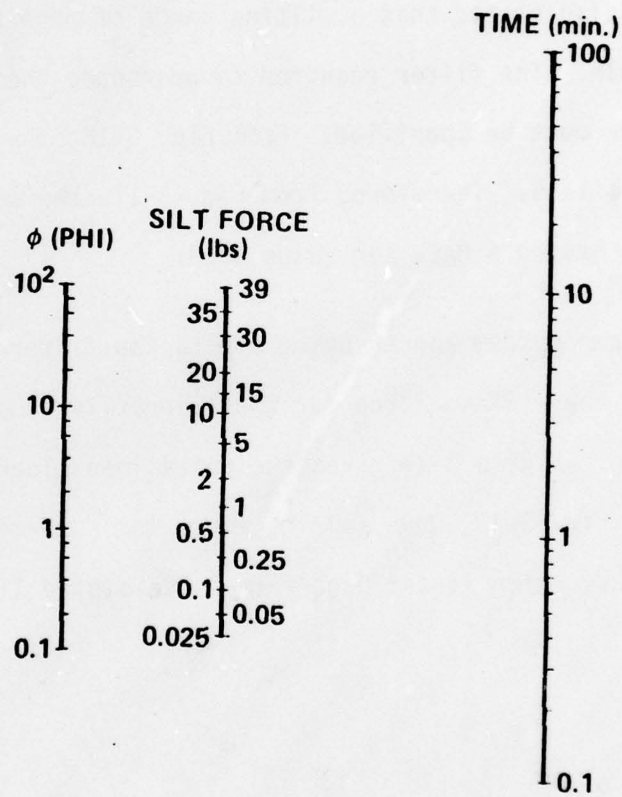


Fig. 3-18. Silting Force Nomograph.

has been determined as 10 in a system with an ingression rate of 10^8. It is stated that the maximum length of inactive time for this valve is to be 10 min. In other words, upon initial movement of the valve spool, the amount of time before any successive movement could be 10 min. It is also stated that a silting force of more than 15 lbs would be intolerable. The filter required to guarantee the safe operation of this valve must be specified. From Fig. 3-18, the contaminant lock coefficient ϕ is 5. Therefore, from Fig. 3-17, the appropriate filter would be one having a Beta ten value of 3.

In the same system above, using a Beta ten filter of 1.1, the magnitude of the silting force for a stationarity time of 10 min can be determined. Figure 3-17 gives the contaminant lock coefficient ϕ of 60. From Fig. 3-18, the silting force for a 10-min stationarity time is 36 lbs, which is far higher than the stated limit of 15 lbs.

CHAPTER IV

VALVE TEST PACKAGE
FOR PROPOSED SPECIFICATION FOR U.S. ARMY

TEST PROCEDURES REQUIRED

To evaluate the integrity of directional-control valves, the following test procedures are required.

1. OSU-V-1. Method of determining and reporting pressure differential-flow characteristics.
2. OSU-V-2. Method of determining and reporting external control characteristics.
3. OSU-V-3. Method for determining metering characteristics.
4. OSU-V-4. Method for determining internal leakage characteristics.
5. OSU-V-9. Method for determining valve durability.
6. OSU-V-10. Method for determining low temperature performance.
7. OSU-V-14. Method for evaluating the structural integrity of the valve.
8. OSU-V-15. Salt spray test method.
9. OSU-V-17. Method of measuring and reporting contaminant lock characteristics.
10. OSU-V-18. Abrasive test.

RATIONALE FOR THE CHANGE OF TEST PROCEDURES IN MIL-V-52688

The test procedures which are cited in MIL-V-52688 and altered by the procedures recommended in this report are listed below with alter-

native test procedures.

OSU No.	Recommended Test Procedure	MIL-V-52688
V-1	Pressure/flow	Pressure drop test
V-2	External control	Spool operating force test
V-3	Metering characteristic	(None)
V-4	Internal leakage	Internal leakage test
V-9	Durability	Endurance test
V-10	Low temperature	Low temperature test
V-14	Structural integrity	Proof pressure test
V-15	Salt spray	Salt spray test
V-17	Contaminant lock	(None)
V-18	Abrasive	Abrasive test

1. For the pressure drop test, OSU-V-1 should be used to greatly reduce energy costs.
2. The spool operating force test in MIL-V-52688 is limited to open-center valves without detent. The test at -25^{o}F cannot be conducted with MIL-2104 fluid because the pour point of MIL-2104 is -15^{o}F. Hence, OSU-V-2 which was endorsed by the Integrity Advisory and Task Force Groups should be used.
3. The metering characteristics test is not cited in MIL-V-52688.
4. The internal leakage test in MIL-V-52688 is vague as to the exact procedure by which the test should be performed. Thus, OSU-V-9 is recommended which minimizes complicated steps of procedure and still achieves a satisfactory endurance test.
5. The endurance test in MIL-V-52688 is complicated without cause. OSU-V-9 is recommended which minimizes complicated steps of procedure and still achieves a satisfactory endurance test.
6. The low temperature test is a difficult test to conduct because of an extremely low operating temperature. Test fluid should be arctic oil and the test temperature should be a reasonable value for field operation. OSU-V-10 is recommended in which

an arctic oil is used at an operating temperature recommended by the Industry Advisory and Task Force Groups.

7. The proof pressure of 3000 psi in MIL-V-52688 is too low for the valve whose rated pressure is close to 3000 psi or above 3000 psi. OSU-V-14 is recommended using a proof pressure of two times the specified rated pressure. This proof pressure (two times the rated pressure) is endorsed by the Industry Advisory and Task Force Groups.
8. The salt spray test in MIL-V-52688 is essentially the same as OSU-V-15 which clearly documents necessary test conditions and test procedure.
9. MIL-V-52688 does not include the contaminant sensitivity test procedure. OSU-V-17 (contaminant lock test) was developed to reinforce the Army valve test procedures.
10. The abrasive test in MIL-V-52688 includes an unrealistic condition due to the absence of the oil. OSU-V-18 is recommended which takes into account the presence of oil circulating through the valve. OSU-V-18 also provides a qualification procedure for an abrasive environment.

TYPICAL RESULTS AND CHARACTERISTIC CURVES

1. Typical results and the characteristic curve of the directional control valve obtained using the OSU-V-1 pressure-flow characteristic test are shown in Table 4-1 and Fig. 4-1.
2. Typical results of OSU-V-2 (external control characteristic test) are illustrated in Figs. 4-2 through 4-5.

TABLE 4-1.

Summary of Actual Pressure Drop-Flow Data
(from Test OSU-V-1).

Flow (Lit/Min)	Valve Pressure: Upstream P_{um} (bars)	Valve Pressure: Downstream P_{dm} (bars)	Valve Pressure: Diff. ΔP_m $(P_{um} - P_{Dm})$	Tare* Pressure Drop ΔP_t (bars)	Pressure Differential ΔP $(\Delta P_m - \Delta P_t)$ (bars)
42.0	2.8	2.4	0.4	0.29	0.11
60.2	4.1	3.5	0.6	0.45	0.15
72.3	5.2	4.4	0.8	0.57	0.23
80.7	5.9	5.0	0.9	0.68	0.22
92.0	7.4	6.0	1.4	0.85	0.55
101.0	9.0	7.2	1.8	1.03	0.77
105.5	9.9	7.9	2.0	1.05	0.95
113.0	11.2	9.0	2.2	1.23	0.97
118.0	12.1	9.7	2.4	1.33	1.07
123.2	13.3	10.6	2.7	1.42	1.28
133.7	15.5	12.1	3.4	1.60	1.80
137.8	16.6	12.8	3.8	1.68	2.12
147.5	18.6	14.6	4.0	1.86	2.14
154.6	20.4	15.9	4.5	2.04	2.46
157.5	24.1	16.55	7.55	2.13	5.42

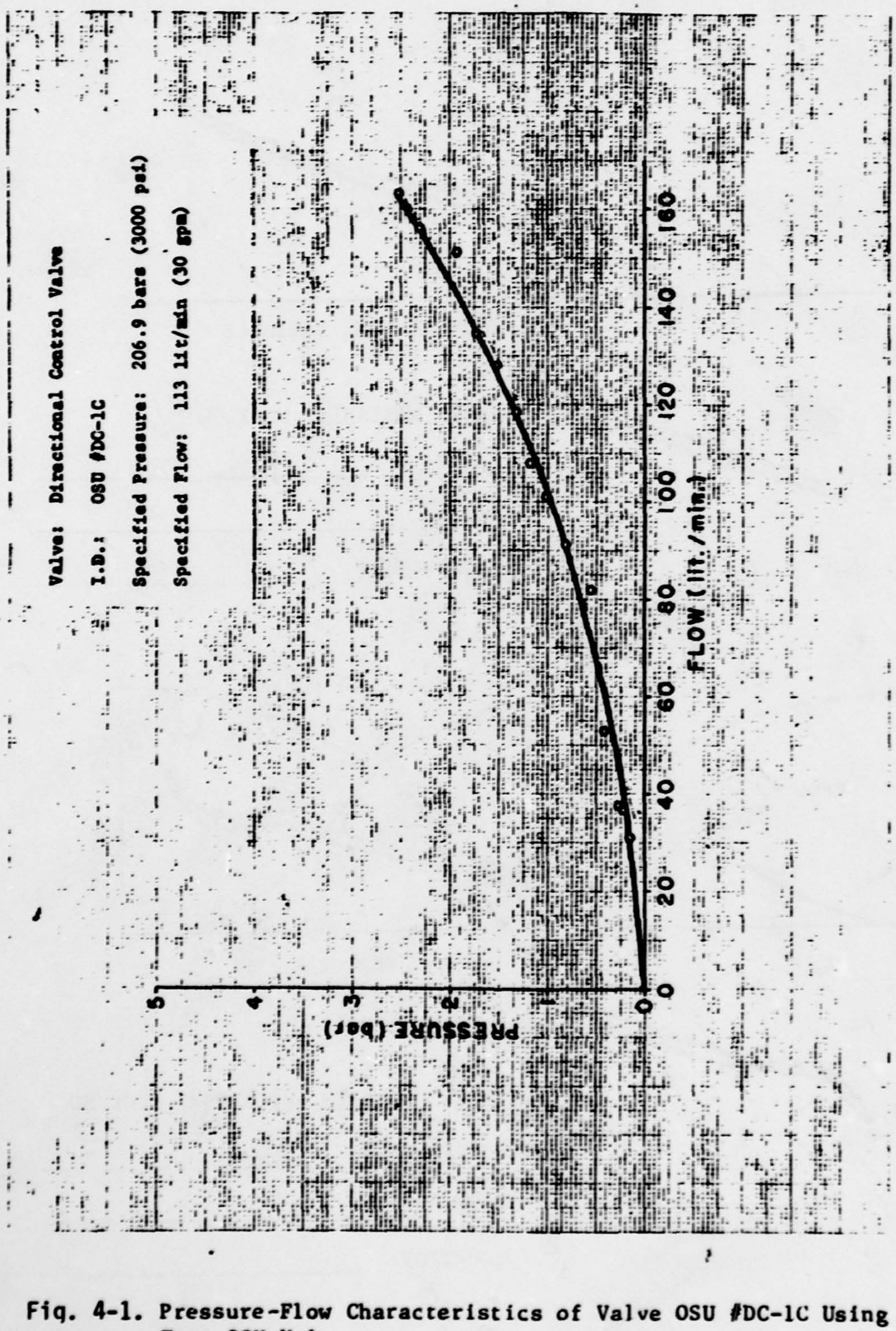


Fig. 4-1. Pressure-Flow Characteristics of Valve OSU #DC-1C Using Test OSU-V-1.

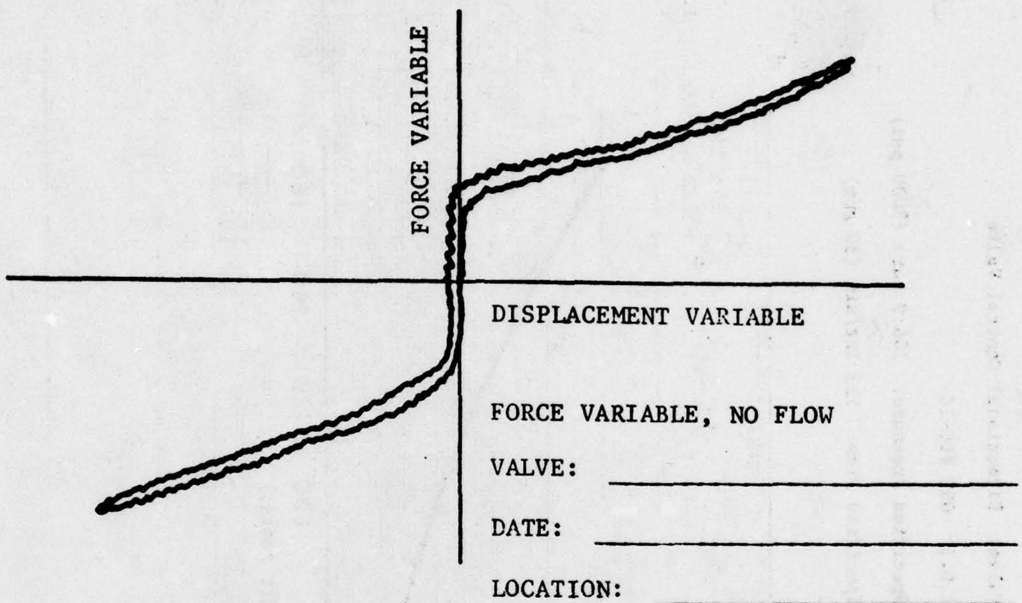

FORCE VARIABLE
DISPLACEMENT VARIABLE
FORCE VARIABLE, NO FLOW
VALVE:
DATE:
LOCATION:

Fig. 4-2.

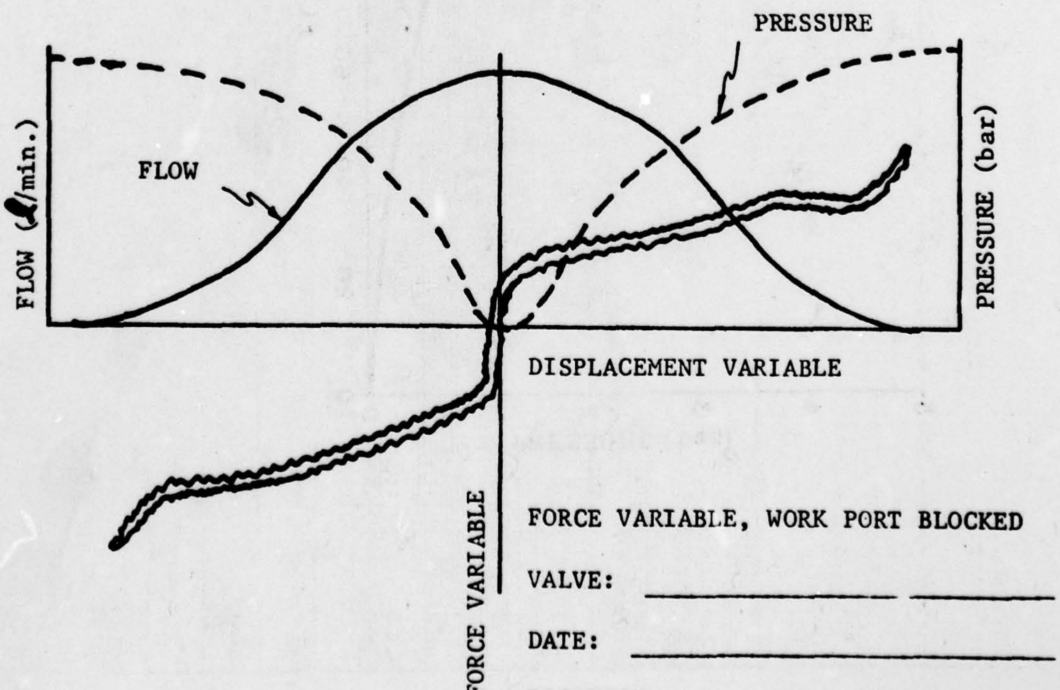
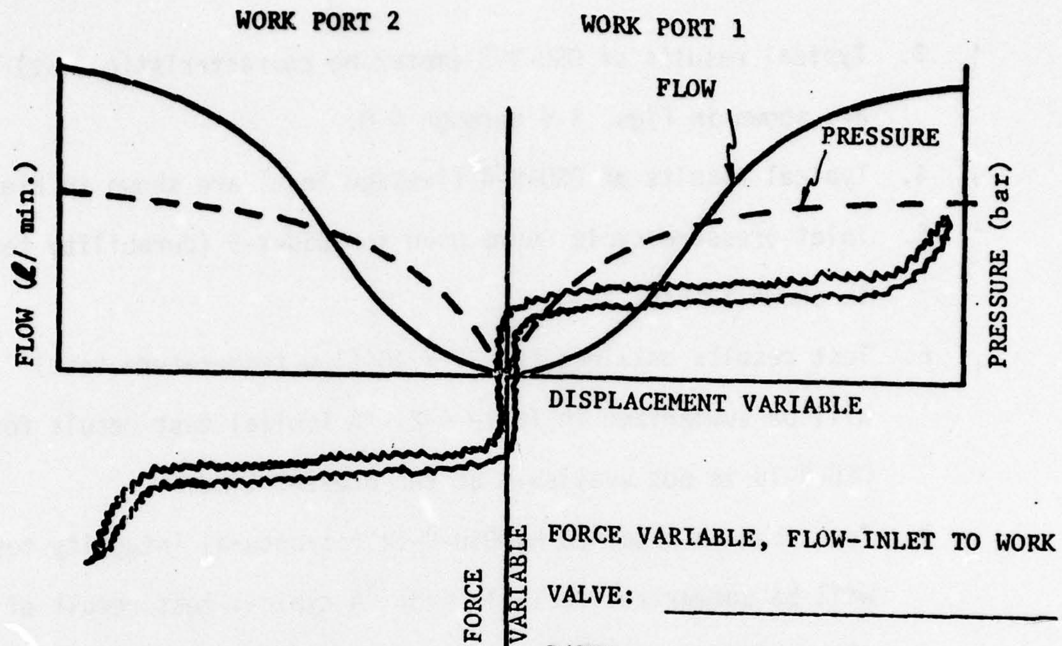

PRESSURE
FLOW (ℓ/min.)
FLOW
PRESSURE (bar)
DISPLACEMENT VARIABLE
FORCE VARIABLE
FORCE VARIABLE, WORK PORT BLOCKED
VALVE:
DATE:
LOCATION:

Fig. 4-3.



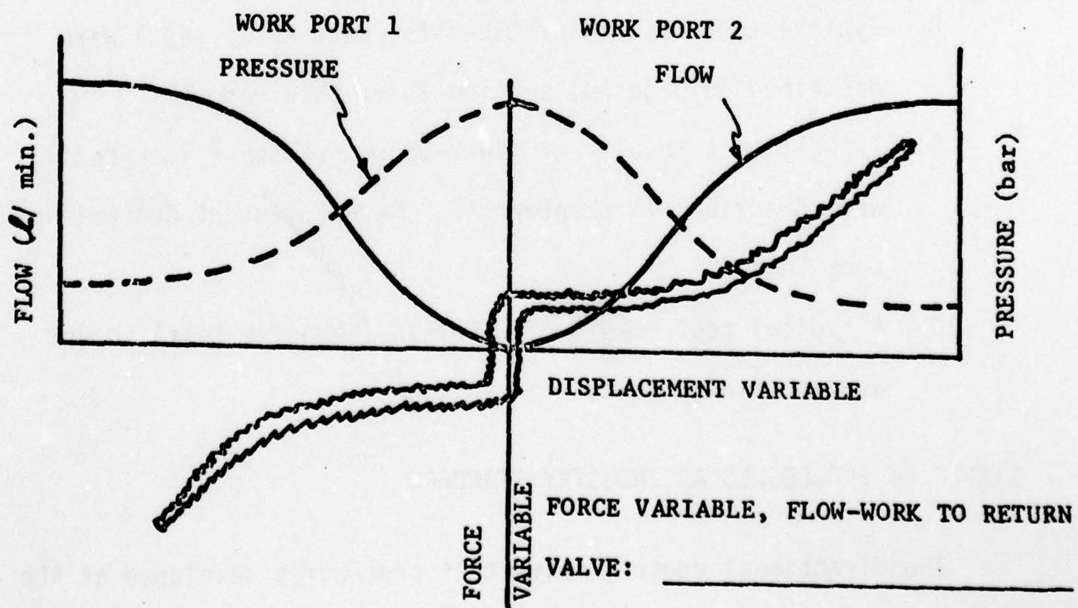
FORCE VARIABLE, FLOW-INLET TO WORK

VALVE: ______________________

DATE: ______________________

LOCATION: ______________________

Fig. 4-4.



FORCE VARIABLE, FLOW-WORK TO RETURN

VALVE: ______________________

DATE: ______________________

LOCATION: ______________________

Fig. 4-5.

3. Typical results of OSU-V-3 (metering characteristic test) are shown in Figs. 4-6 through 4-8.
4. Typical results of OSU-V-4 (leakage test) are shown in Fig. 4-9.
5. Inlet pressure cycle curve used for OSU-V-9 (durability test) is shown in Fig. 4-10.
6. Test results obtained by OSU-V-10 (low temperature test) will be summarized in Table 4-2. A typical test result for OSU-V-10 is not available at the present time.
7. Test results obtained by OSU-V-14 (structural integrity test) will be summarized in Table 4-3. A typical test result of OSU-V-14 is also not available at the time of this publication.
8. Typical test results of OSU-V-15 (salt spray test) were described on page 10, section 7, of this report.
9. Typical test results of OSU-V-17 (contaminant lock test) were described in Chapter III, "Development of Contaminant Lock Test."
10. A typical test result of OSU-V-18 (abrasive test) is not available at present.

STATUS OF PROCEDURES AS INDUSTRY STANDARD

The directional control valve test procedures developed at the Fluid Power Research Center, Oklahoma State University, have been submitted to the SAE and NFPA to be considered as industrially-accepted test procedures. The present status of the procedures are summarized as follows.

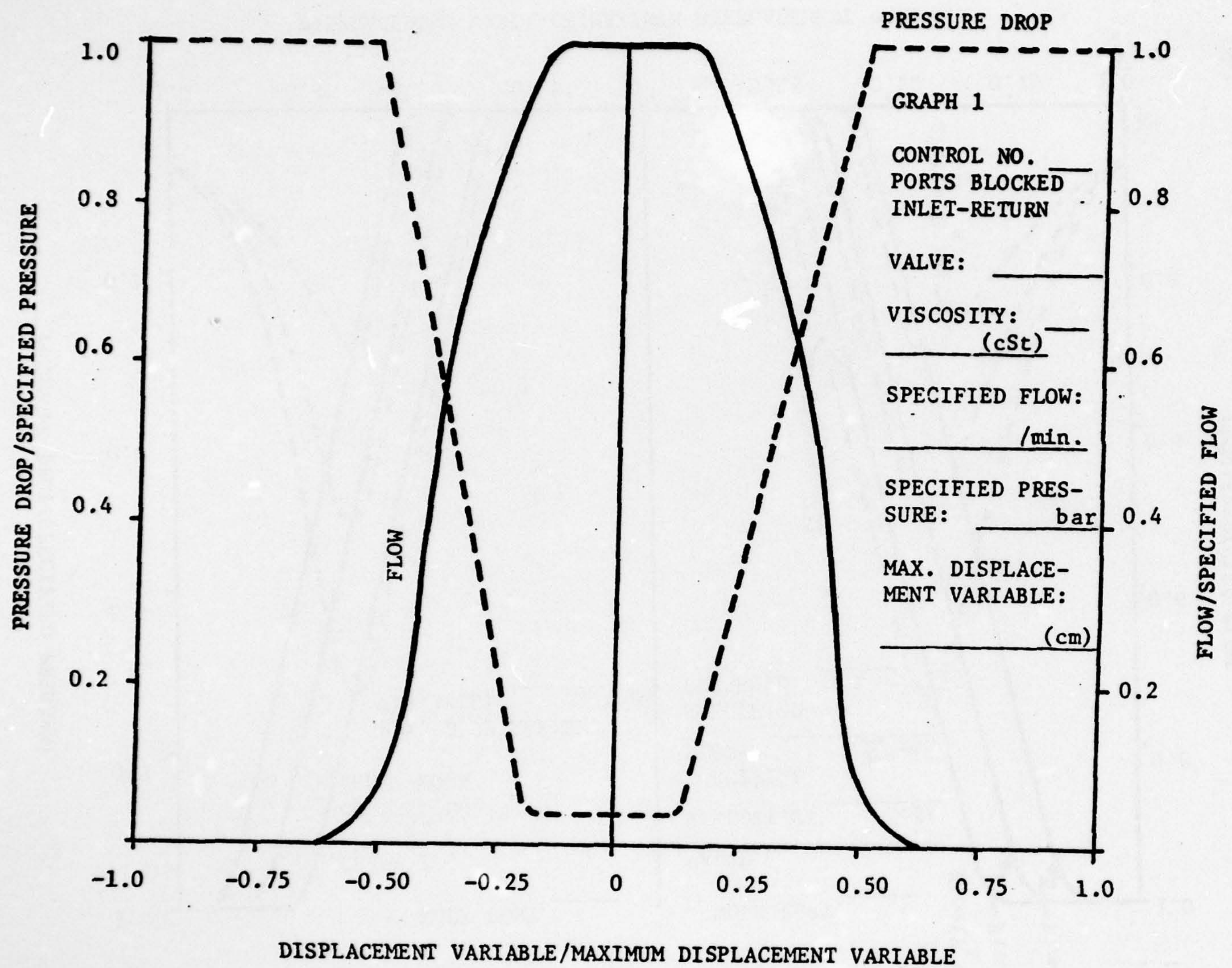
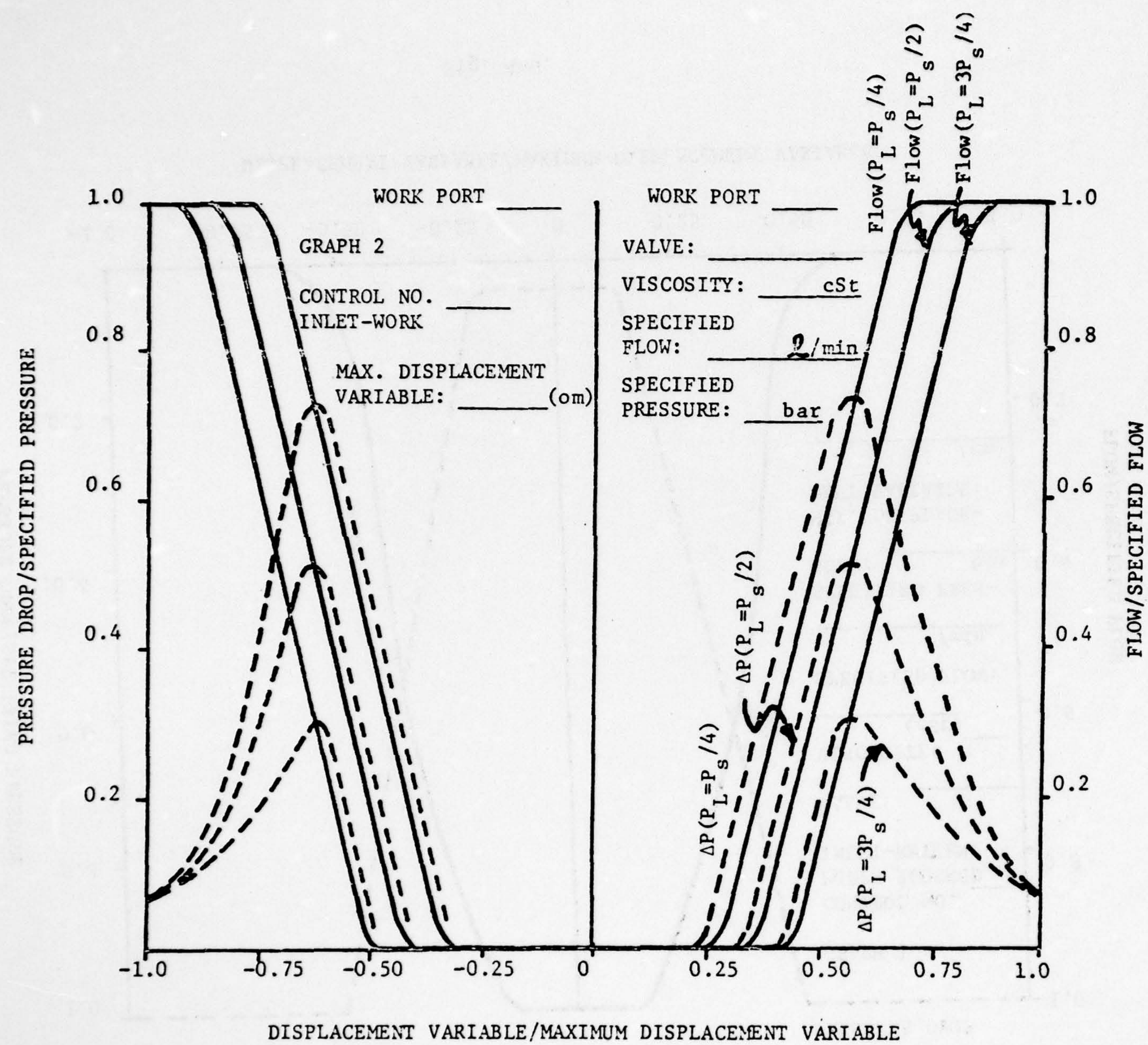

PRESSURE DROP
GRAPH 1
CONTROL NO.
PORTS BLOCKED
INLET-RETURN
VALVE:
VISCOSITY:
(cSt)
SPECIFIED FLOW:
/min.
SPECIFIED PRES-
SURE: bar
MAX. DISPLACE-
MENT VARIABLE:
(cm)
FLOW
1.0
0.8
0.6
0.4
0.2
-1.0
-0.75
-0.50
-0.25
0
0.25
0.50
0.75
1.0
PRESSURE DROP/SPECIFIED PRESSURE
FLOW/SPECIFIED FLOW
DISPLACEMENT VARIABLE/MAXIMUM DISPLACEMENT VARIABLE

Fig. 4-6.



WORK PORT
WORK PORT
GRAPH 2
CONTROL NO.
INLET-WORK
MAX. DISPLACEMENT
VARIABLE: (om)
VALVE:
VISCOSITY: cSt
SPECIFIED
FLOW: ℓ/min
SPECIFIED
PRESSURE: bar
Flow(P_L=P_s/4)
Flow(P_L=P_s/2)
Flow(P_L=3P_s/4)
ΔP(P_L=P_s/4)
ΔP(P_L=P_s/2)
ΔP(P_L=3P_s/4)
PRESSURE DROP/SPECIFIED PRESSURE
FLOW/SPECIFIED FLOW
1.0
0.8
0.6
0.4
0.2
-1.0
-0.75
-0.50
-0.25
0
0.25
0.50
0.75
1.0
DISPLACEMENT VARIABLE/MAXIMUM DISPLACEMENT VARIABLE

Fig. 4-7.

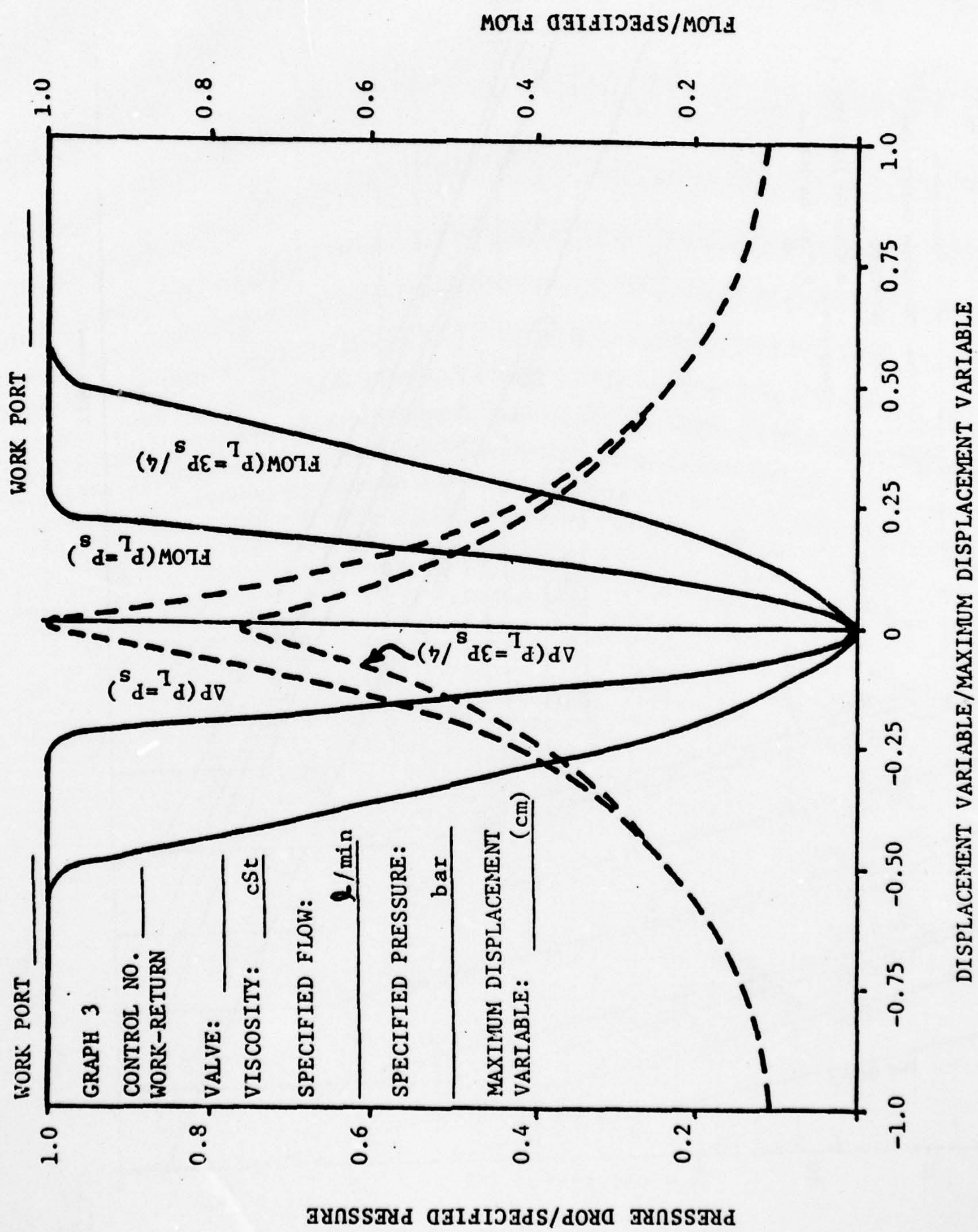

WORK PORT
WORK PORT
GRAPH 3
CONTROL NO.
WORK-RETURN
VALVE:
VISCOSITY: cSt
SPECIFIED FLOW: ℓ/min
SPECIFIED PRESSURE: bar
MAXIMUM DISPLACEMENT VARIABLE: (cm)
FLOW($P_L=P_s$)
FLOW($P_L=3P_s/4$)
ΔP($P_L=P_s$)
ΔP($P_L=3P_s/4$)
PRESSURE DROP/SPECIFIED PRESSURE
FLOW/SPECIFIED FLOW
DISPLACEMENT VARIABLE/MAXIMUM DISPLACEMENT VARIABLE
1.0
0.8
0.6
0.4
0.2
-1.0
-0.75
-0.50
-0.25
0
0.25
0.50
0.75
1.0

Fig. 4-8.

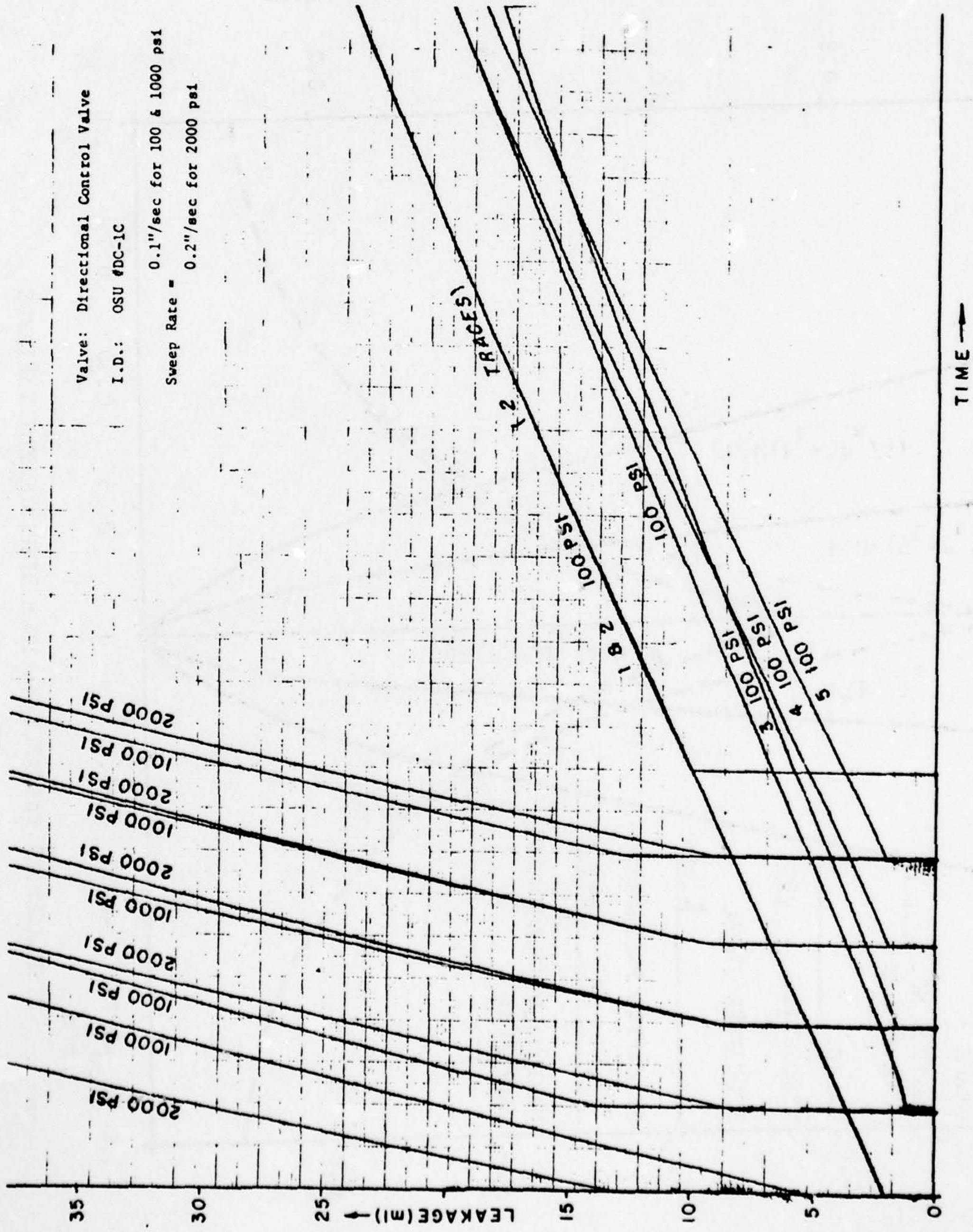


Fig. 4-9. Leakage Characteristics of OSU #DC-1C Valve Using Test OSU-V-4.

AD-A073 789 OKLAHOMA STATE UNIV STILLWATER FLUID POWER RESEARCH --ETC F/G 13/7
MERADCOM/OSU HYDRAULIC SYSTEM RELIABILITY PROGRAM.(U)
MAY 79 DAAK70-77-C-0097

UNCLASSIFIED OSU-FPRC-9MI NL

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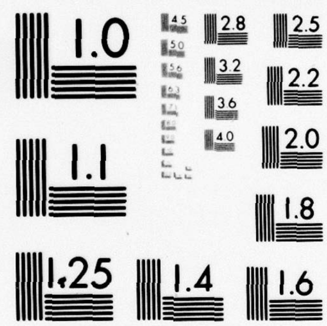


MICROCOPY RESOLUTION TEST CHART

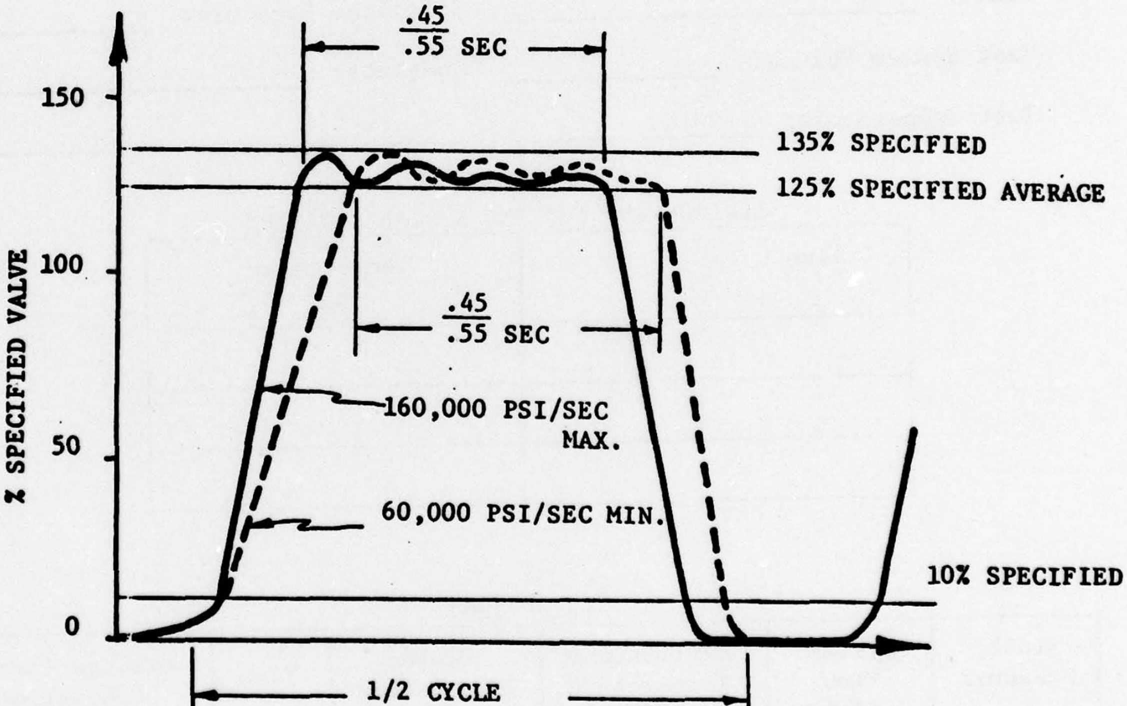


Fig. 4-10. Inlet Pressure Cycle Associated With Durability Test.

TABLE 4-2.

EXAMPLE LOW TEMPERATURE TEST DATA

Date Tested: ____________ Test Location: ____________

Valve: ____________ Specified Flow: ____________

Fluid: ____________ Specified Pressure: ____________

Test System Volume: ____________ Comments: ____________

Test Temperature: -30°C. ____________

ENVIRONMENT COOLING & STABILIZATION

Time	Temperature °C

VALVE TEST

Inlet Pressure (bar)	System Flow (ℓ/min)	Environment Temperature °C	Fluid Temperature °C	Time	Leakage (ℓ/min) & Location

Mechanically Operated
Directional Control Valve

1. Time to Return to Neutral After Activation: ____________(sec)
(Fluid Temperature -15°C ± 6°C.)
2. Force Required to "Recenter" After: ____________(N)
One Minute Waiting Period

TABLE 4-3.

EXAMPLE TEST DATA SUMMARY STRUCTURAL INTEGRITY TEST

Test Date: ____________________ Test Location: ____________________

Valve: ____________________ Fluid: ____________________

Specified Input Pressure: ______ (bar) Fluid Temperature: ______ (^{o}C)

Specified Output Pressure: ______ (bar) Viscosity: ______ (cSt)

Specified Work Port Pressure: ______ (bar) Comments: ____________________

Port Type and Location	Test Pressure (bar)	Leakage Rate (ℓ/min) and Location

OSU No.	Procedure	Document Number SAE	NFPA
V-1	Pressure/ flow	J1117	T3.5.28-1977
V-2	External control	XV-1 (Being considered as SAE procedure)	T3.5.13-
V-3	Metering character- istic		T3.5.14-(Second balloting in April, 1979)
V-4	Internal leakage	J1235	T3.5.15-(Second general review in April, 1979)
V-9	Durability	Submitted in February, 1979	T3.5.20-
V-10	Low tem- perature	Submitted in February, 1979	T3.5.21-
V-14	Structural integrity	Submitted in February, 1979	T3.5.25-
V-15	Salt spray	Submitted in May, 1979	
V-17	Contaminant lock	Submitted in February, 1979	
V-18	Abrasive	Submitted in May, 1979	

Valve test procedures presented herein which are not contained in current SAE or NFPA standards are presented in Appendix A.

CHAPTER V

DRAFT OF PROPOSED ARMY TEST PROCEDURES

SCHEDULE OF TESTS

Some of the test procedures proposed in this report are destructive tests shown as follows.

OSU No.	Procedure	Destructive?
V-1	Pressure/flow	No
V-2	External control	No
V-3	Metering characteristic	No
V-4	Internal leakage	No
V-9	Durability	Yes
V-10	Low temperature	Occasionally
V-14	Structural integrity	Occasionally
V-15	Salt spray	Yes
V-17	Contaminant lock	Occasionally
V-18	Abrasive	Yes

Hence, four (or five) identical test valves are necessary to achieve the complete test package. The following test sequence shall be followed to complete all the tests.

Valve 1 → Pressure/Flow → External Control → Metering Characteristic → Internal Leakage → Low Temperature → Salt Spray

Valve 2 → Structural Integrity → Valve is still OK? → No → Valve 5 → Abrasive

Valve is still OK? → Yes → Abrasive

Valve 3 → Durability

Valve 4 → Contaminant Lock

ACCEPTABLE VALUES

At the present time, it is difficult for most of the procedures to determine exact values that will be used as test evaluation criteria. This is becuase few test data have been obtained to determine acceptable valves. A practical method of selecting the best valve is to test several candidates and take the best. The following table shows they way to select the best valve for each test procedure.

OSU No.	Procedure	Criterion
V-1	Pressure/flow	Minimum pressure differential
V-2	External control	Linearity of input-output relationship (depends on application)
V-3	Metering characteristic	Wide control range and linearity
V-4	Internal leakage	Minimum leakage
V-9	Durability	Should pass all criteria
V-10	Low temperature	Minimum change of performance parameters due to low temperature
V-14	Structural integrity	Minimum change of leakage rate due to proof pressure
V-15	Salt spray	No effect of salt spray
V-17	Contaminant lock	Minimum Omega valve rating
V-18	Abrasive	Minimum change of pressure/flow characteristic and internal leakage

CHAPTER VI

DISCUSSION OF ADDITIONAL SPECIFICATION ITEMS

Other specification items—valve noise generation and performance modeling—have been studied. A preliminary experiment was conducted for the valve noise generation, and the performance modeling technique was studied. Further studies are necessary to formalize the test procedures for these two specifications. The effort expended to study these two specifications in this project is reported in the following paragraphs.

VALVE NOISE GENERATION POTENTIAL

The obtrusive noise generated by hydraulic components has gradually attracted more and more complaints by those concerned with hydraulic units. Since pumps and motors are the major noise generators in hydraulic systems, much effort has been expended in developing quieter pumps and motors.

Since the development of quieter pumps and motors, valve noise has now become a problem to contend with. The valve noise level was measured as a function of pressure differential across the valve. The valve noise level was then related to the Reynolds number which describes the degree of turbulence in flow. An attenuation of the valve noise by increasing back pressure was also tested and evaluated.

1. Qualification of Test Procedure

Since the relative noise level measurement method was employed to evaluate the valve noise level, a correct evaluation of the valve should be qualified by checking the noise level difference between the valve and the environment. A sound pressure level difference of 6 dB or more will validate the valve noise measurement.

Before starting the valve noise measurement, sound pressure levels were measured at 0.4 inches, 10 inches, 20 inches, and 40 inches from the valve while in operation. The sound pressure level at the distance of 40 inches from the valve was considered as the background noise level. Fig. 6-1 shows a result of the measurements at different distances from the valve. A sound pressure level difference of 6 dB between the measurements at 0.4 inches and 40 inches validates the correct measurement of valve noise at 0.4 inches.

2. Valve Noise Vs Reynolds Number

When pressure differential across the valve increases, the turbulence of fluid flow is increased and the noise level elevated. Since the degree of turbulence in fluid flow is usually described by a Reynolds number, it is more significant to express the valve level as a function of Reynolds number rather than a function of pressure differential. Thus, the valve noise level vs Reynolds number characteristics can be used as a common tool in the selection of the quietest valve regardless of the valve shape and size.

Fig. 6-2 shows the valve noise level vs Reynolds number characteristic of OSU Valve No. 103. The sound pressure level of OSU Valve

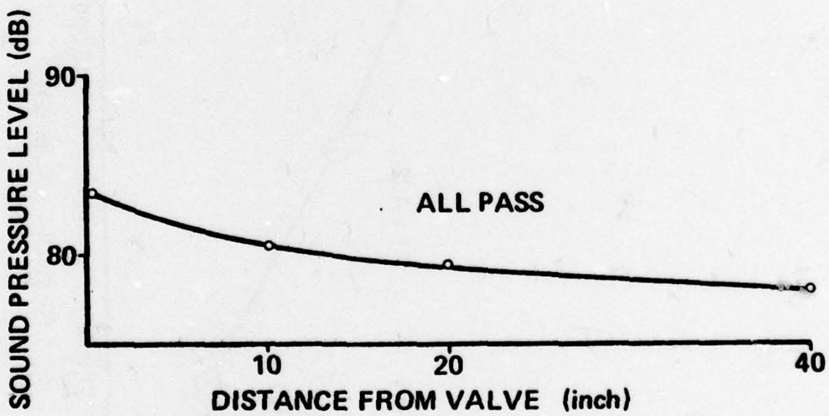


Fig. 6-1. Qualification of Valve Noise Measurement

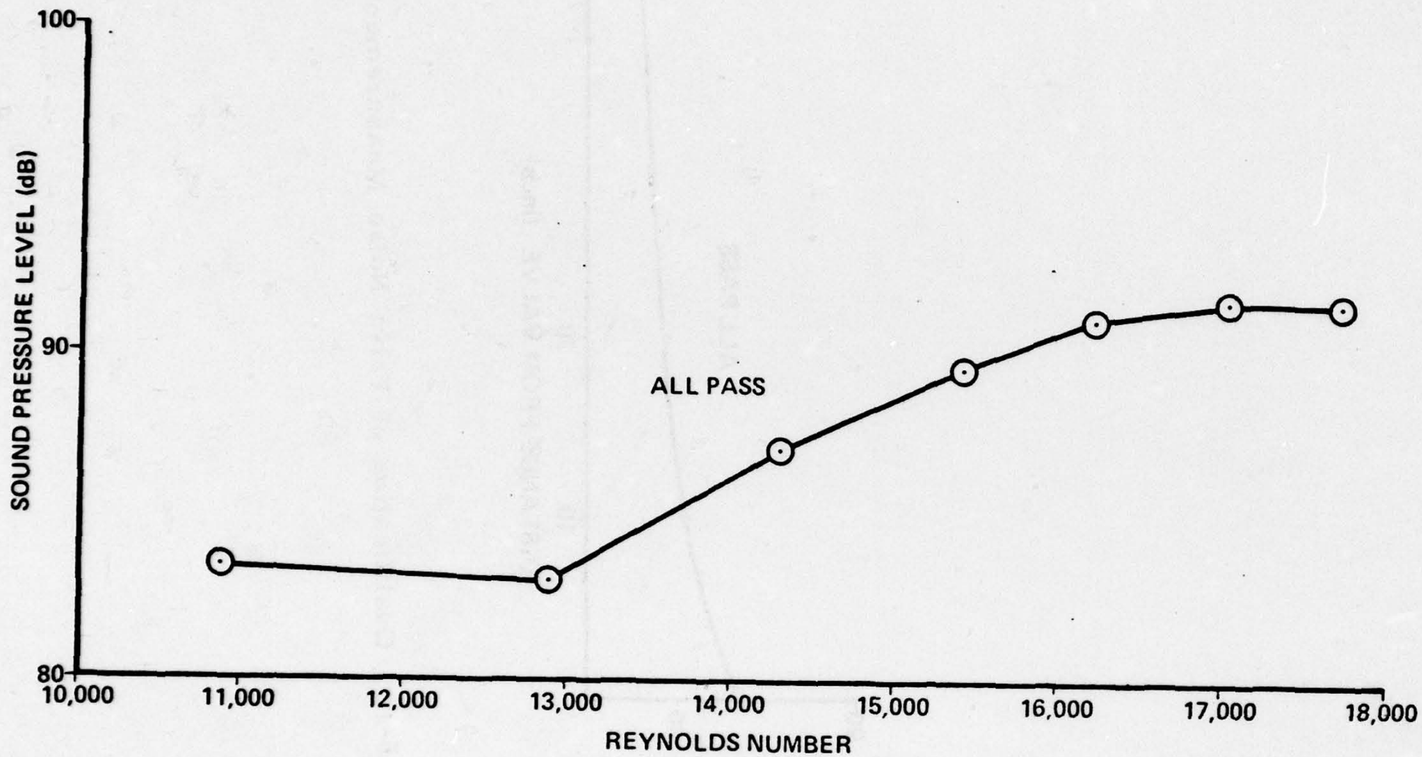


Fig. 6-2. Valve Noise Level Versus Reynolds Number

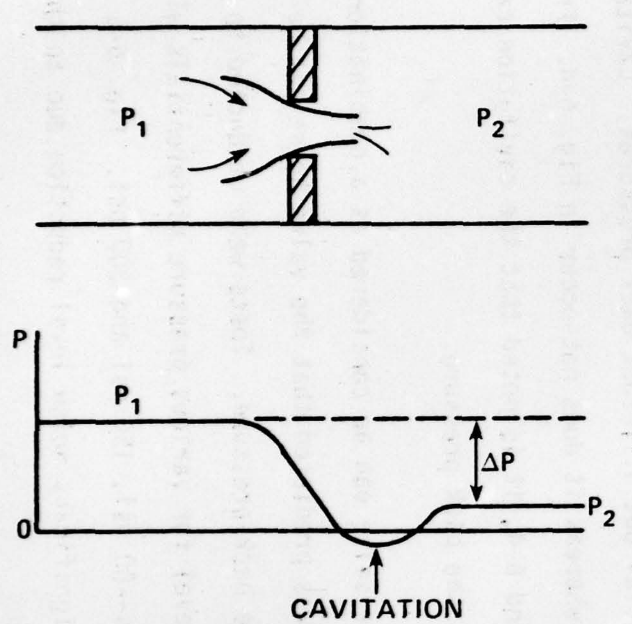
No. 103 was measured first for various pressure differentials. Then, Reynolds numbers associated with measured pressure differentials were calculated using valve dimensions so that the sound pressure level vs Reynolds number characteristics could be formalized.

In Fig. 6-2, no increase of the noise level is observed in the range of Reynolds number less than 13,000. From Re 13,000 to Re 17,000, an obvious increase of thenoise level is seen; however, this also levels off and becomes a constant noise level at Re >17,000.

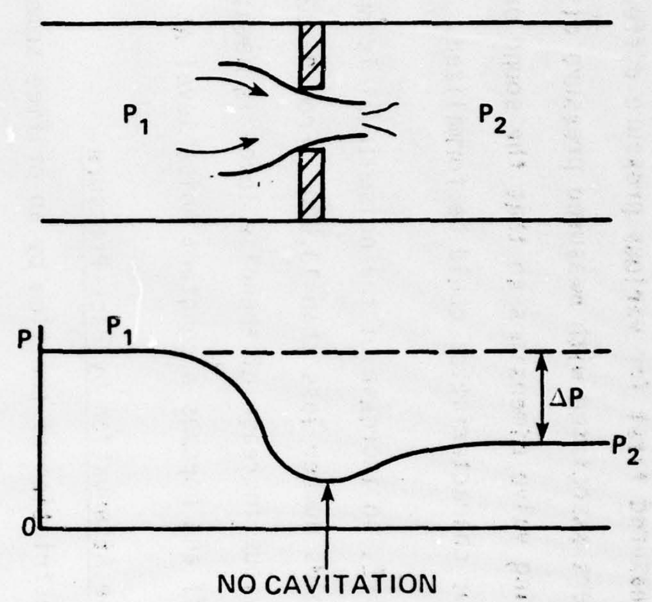
3. Valve Noise Attenuation by Back Pressure

Sudden restriction of the flow by an orifice sometimes causes cavitation. When cavitation occurs in the orifice region, a significant increase in the noise level is observed. It is known that the cavitation tends to take place when the pressure downstream of the orifice becomes less than atmospheric pressure. Fig. 6-3 and 6-4 show pressure distributions in the orifice region with the same pressure differential but different back pressures. Cavitation occurs in Fig. 6-3, whereas it does not occur in Fig. 6-4. From the comparison of Fig. 6-3 and 6-4, it is noted that the cavitation can be prevented by increasing the back pressure.

Since the valve can be considered as a combination of various orifices, it was predicted that the valve noise can be attenuated by increasing the back pressure. Tests were conducted to measure the valve noise level for various pressure differentials with different back pressures—80 psi, 150 psi and 300 psi. Fig. 6-5 shows the test results. A significant noise level reduction due to the increased



Orifice Flow With Cavitation
Fig. 6-3



Orifice Flow Without Cavitation
Fig. 6-4

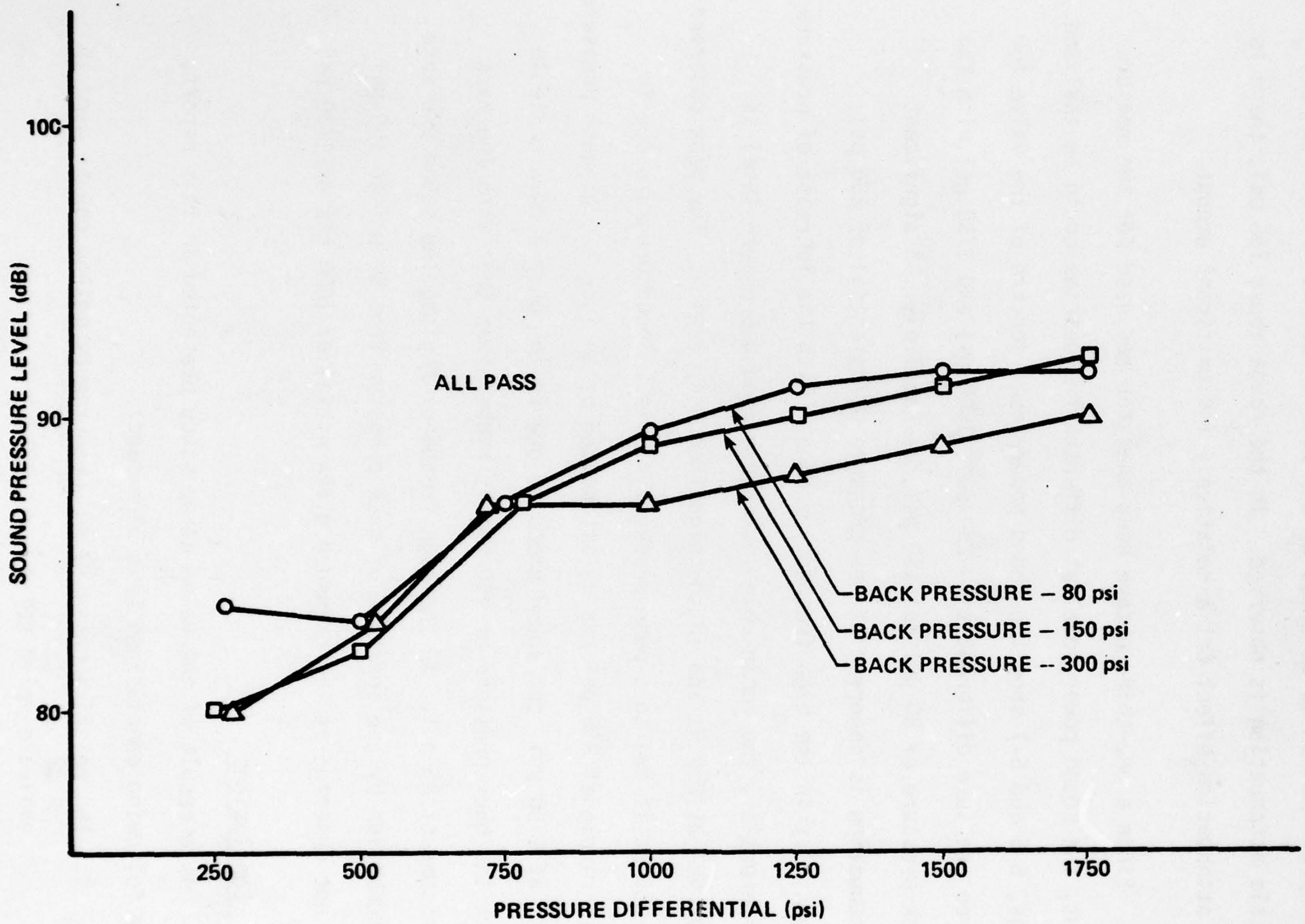


Fig. 6-5. Valve Noise Attenuation By Back Pressure

back pressure can be observed at 250 psi. From 500 psi to 750 psi, little attenuation is observed. In the range above 750 psi, there is an attenuation effect but a relatively insignificant amount.

Since a one-third octave band analyzer was used for the measurement, the sound power level at different frequencies could be analyzed. Figs. 6-6 and 6-7 show the sound power level spectra of the valve for three pressure differentials—250 psi, 1000 psi and 1750 psi with the back pressure of 80 psi and 150 psi, respectively. A signficant attenuation is observed at the pressure differential of 250 psi, expecially in the high frequency region. With the increase of pressure differential, the remarkable increase of sound pressure level is observed at 250 Hz and in the high frequency region. The peak observed at 250 Hz is due to a pump pressure ripple. Sound pressure due to this ripple at 250 psi can be attenuated by an increase of back pressure. But, at 1750 psi, the sound pressure due to the pump pressure ripple with the back pressure of 150 psi is higher than that with the back pressure of 80 psi. In the high frequency region, the sound pressure attenuation by the increase of back pressure from 80 psi to 150 psi is not observed at the pressure differential of 1000 psi or 1750 psi.

4. Conclusions

As a result of the valve noise study presented in this report, the following conclusions have been made:

1. The relative noise level measurement method can be used to evaluate valve noise.

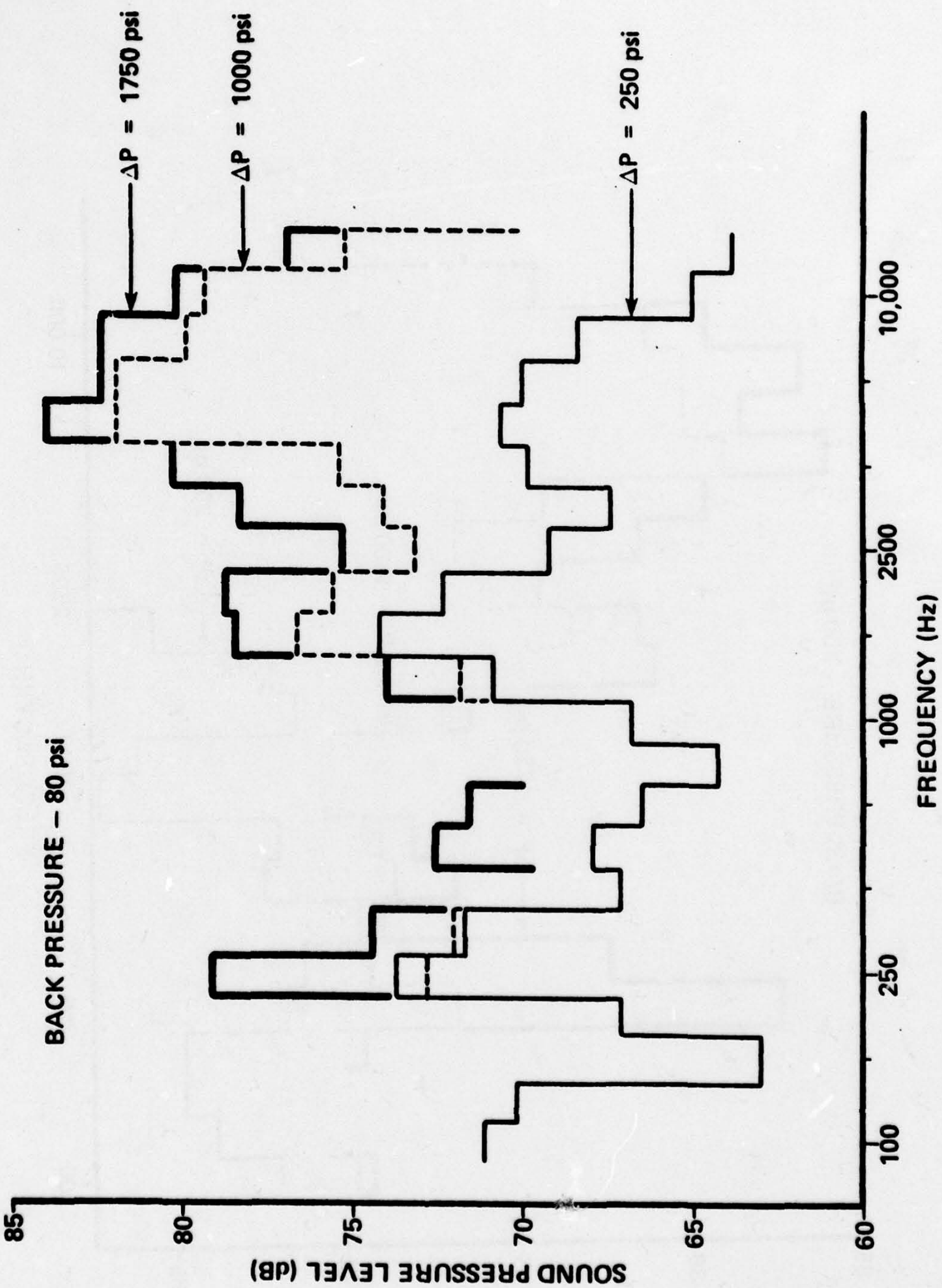


Fig. 6-6. Valve Sound Pressure Level Spectra with Back Pressure of 80 psi.

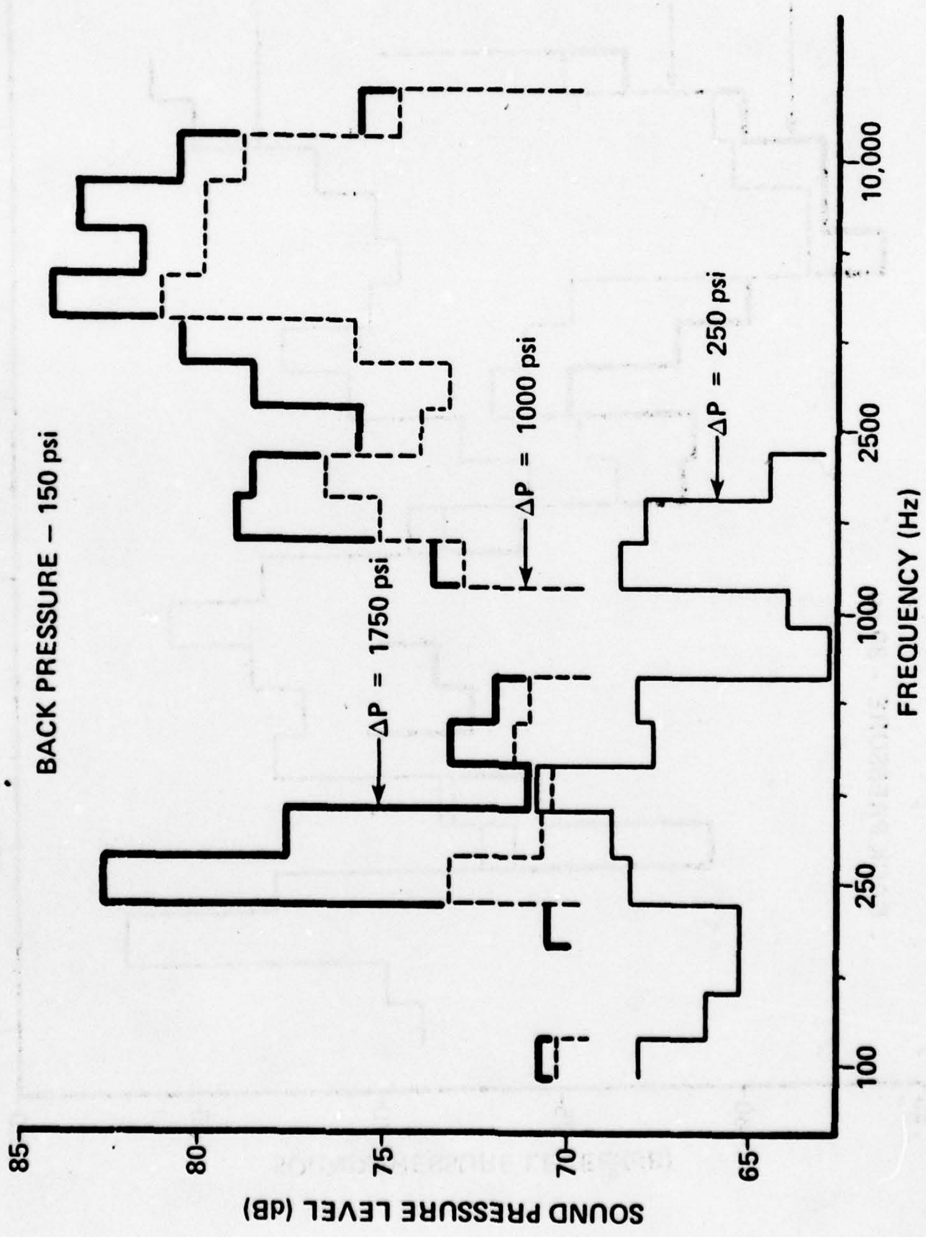


Fig. 6-7. Valve Sound Pressure Level Spectra with Back Pressure of 150 psi.

2. The test valve shows the noise level increase in the region of Reynolds number 13,000 to 17,000. Above Reynolds number 17,000, the noise level is stabilized.
3. The valve noise level attenuation due to the increase of back pressure from 80 psi to 150 psi or 300 psi is significant at the pressure differential across the valve of 250 psi. But, it is not significant in the range of the pressure differential above 500 psi.
4. The sound pressure due to the pump pressure ripple existing at a frequency of 250 Hz is attenuated by increasing the back pressure from 80 psi to 150 psi when the pressure differential across the valve is 250 psi. But, when the pressure differential is 1000 psi or 1750 psi, no attenuation is observed.
5. The sound pressure level above the frequency 1000 Hz is significantly attenuated by increasing the back pressure from 80 psi to 150 psi when the pressure differential across the valve is 250 psi. But, when the pressure differential is 1000 psi or 1750 psi, no significant attenuation of the sound power level is observed in the the entire frequency range.

PERFORMANCE MODEL TECHNIQUE

1. Gray Box Method

The classical method which has been exclusively used to model the system is called the "black box method." To model the system by the black box method, the following procedure should be followed.

1. Isolate the component from the rest of the system and decide which quantities are to be treated as inputs, outputs and disturbances.
2. Idealize and lump all elements involved.
3. By applying the laws of mechanics, formulate the mathematical relationship between variables and parameters.
4. Impose inputs to the model so that its behavior can be studied.

Through the above mentioned procedure, the system model is derived and verified. However, the black box method has the following disadvantages.

1. A large number of parameters are involved in the modeling process, and therefore, manipulation of the parameters is usually complicated.
2. Interfacing such a model with other components is difficult because there is no common variable that can be used as the input variable or the output variable.
3. Nonlinearity of the system causes problems because the black box method is essentially employed to model linear systems.
4. Few state variables are available to measurement, hindering verification of the model.

In order to overcome these disadvantages of the black box method, a "gray box method" has been developed at the Fluid Power Research Center. The gray box method is a general time domain analysis technique developed by using the multi-port concept which has also been developed at the Fluid Power Research Center. In modeling a process

by the gray box method, only through and across variables which can be easily measured are of concern. The gray box method is called a semi-empirical method because the parameters involved in the modeling are experimentally identified. The model derived by the gray box method is not design-dependent since no geometric parameters are involved. Therefore, similar components can be easily compared with respect to their performances.

2. Power Port Model

The product of the through and across variable is the power. Since the through and across variables are considered as the input and output variables in the gray box method, the power is one variable which can interconnect components. The power port model concept is used to optimize a total system by interconnecting the gray box model of each component using the power as a common variable. Since the power is the only variable which should be optimized, the optimization process can be greatly simplified. The power port model improves design trade-off studies in that the parameters of the gray box model can be replaced in the optimization process to evaluate the system responses of any alternative component.

CHAPTER VII

RECOMMENDATIONS AND CONCLUSIONS

The primary objective of this project was to evaluate and improve the performance requirements of directional control valves described in MIL-V-52688. The tests were conducted by the procedures in MIL-V-52688 and the results were presented in Chapter II. As a result of the investigation, all test procedures in MIL-V-52688 were improved. Some of them have already been approved as SAE or NFPA recommended test procedures. Others were submitted to the SAE and NFPA and are being considered as their standard procedures.

In addition, two new test procedures were developed; they are the metering characteristic test and the contaminant lock test. As the operating pressure of fluid power systems tend to increase and leakage clearances decrease, in order to improve valve efficiency, contaminant lock becomes a major problem in spool-type directional-control valves. The contaminant lock test procedure for spool-type directional-control valves has been developed with its interpretation technique in this project and submitted to the SAE subcommitted for approval.

Among the procedures in MIL-V-52688, the low temperature test was an extremely difficult test because of its unrealistically low temperature with MIL-2104 fluid, and therefore it was replaced with OSU-V-10 (low temperature characteristic test) which has also been submitted to the SAE subcommittee.

Enough test data from the contaminant lock test have been generated in the Fluid Power Research Center to allow the test procedure to be easily accepted in industry. Based on this fact, it is strongly recommended to produce as much supporting test data for the other test procedures as possible in order to make them appealing for industrial acceptance. With much test data available, we will be able to determine acceptable values for the tests which would be accepted in industry.

Two other performance specifications—noise limit and performance modeling—are still under development. However, the initial investigations clarified the possibility of formalizing these specificiations. Test procedures for these items will be formulated as a continuation of the initial investigation upon completion of further study.

REFERENCE

1. Inoue, R., and J. Gillum, "Contaminant Lock in Spool Type Directional Control Valves—Parts 1, 2, 3, 4, and 5," The BFPR Journal, 1980, 13: 163-188.

APPENDIX A

Valve Test Procedures
Presented Which Are Not
Contained In Current
SAE or NFPA Standards

METHOD FOR EVALUATING THE DURABILITY OF

A HYDRAULIC FLUID POWER VALVE - SAE XJXXX

1. Purpose - To provide a uniform testing and reporting method for determining that a hydraulic fluid power valve has a minimum service life under specified test conditions.

2. Terms and Definitions - (For definitions of terms not herein defined, see Ref. [1].).

2.1 Specified Flow - Steady-state flow rate for the component as specified by the manufacturer.

2.2 Specified Pressure - Steady-state operating pressure for the component as specified by the manufacturer.

2.3 Control Variable - Any variable which causes the controlled output characteristic of the valve to change.

2.4 Specified Cycles - The number of test cycles specified by the manufacturer.

3. Units

3.1 The International System of Units (SI) is used herein in accordance with Ref. [2].

3.2 Approximate conversion to SI units appear in parentheses after their "Customary U.S." counterpart.

4. Graphic Symbols - Graphic symbols used herein are in accordance with Refs. [3], [4], and [5]. Where References [4] and [5] are not in agreement with [3], Ref. [3] governs.

5. Summary of Designated Information.

5.1 Specify the following information on all requests for this test.

5.1.1 A description of valve.

5.1.2 A description of fluid (if different from paragraph 10.1).

5.1.3 The fluid temperature (if different from the standardized value in paragraph 10.2).

5.1.4 Test pressure.

5.1.5 Test flow rate.

5.1.6 The input port(s).

5.1.7 The output port(s).

5.1.8 The work port(s).

5.1.9 The control variable.

6. General Procedure.

6.1 Conduct the test in accordance with the fixed values specified by the test request.

6.2 Use only standardized valves, shown in paragraph 10.0, for catalog information and sales literature.

6.3 Test and report of subplate mounted valves may be run with the subplate included. Reports shall explicitly indicate how such valves were tested.

7. Test Conditions

7.1 Accuracy - Maintain the test condition accuracy within the limits shown in the following table:

Test Condition	Maintain Within ±
Flow	2%
Pressure	2%
Temperature	$3^{o}C(5^{o}F)$
Force	5%
Mechanical Displacement	2%
Voltage	3%
Current	3%

7.2 Contaminant Level - Limit the number of particles in the system fluid to a maximum of 1000 particles per millilitre greater than 10μm.

8. Test Procedure

8.1 Install the valve in the test circuit (Reference Fig. A-1).

8.2 Operate the valve of 100% of the specified cycles.

8.3 Operate each valve control variable throughout its entire range during each input cycle.

8.4 Regulate the output of the valve (flow or pressure) during each pressure or flow cycle to attain an average inlet pressure level of at least 125% (not to exceed 135%) of the specified valve for a minimum of 45% of the cycle and a maximum of 55% of the cycle. (See Fig. A-2).

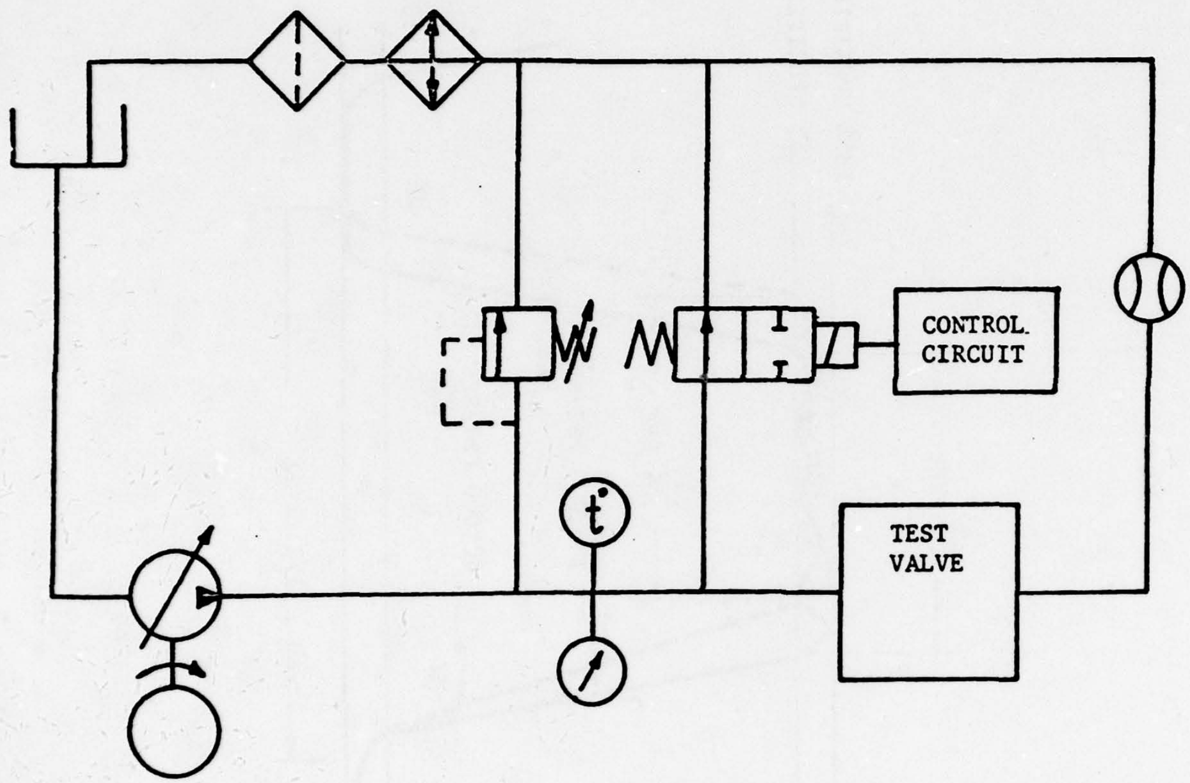


Fig. A-1. Schematic Example of a Hydraulic Valve Durability Test System.

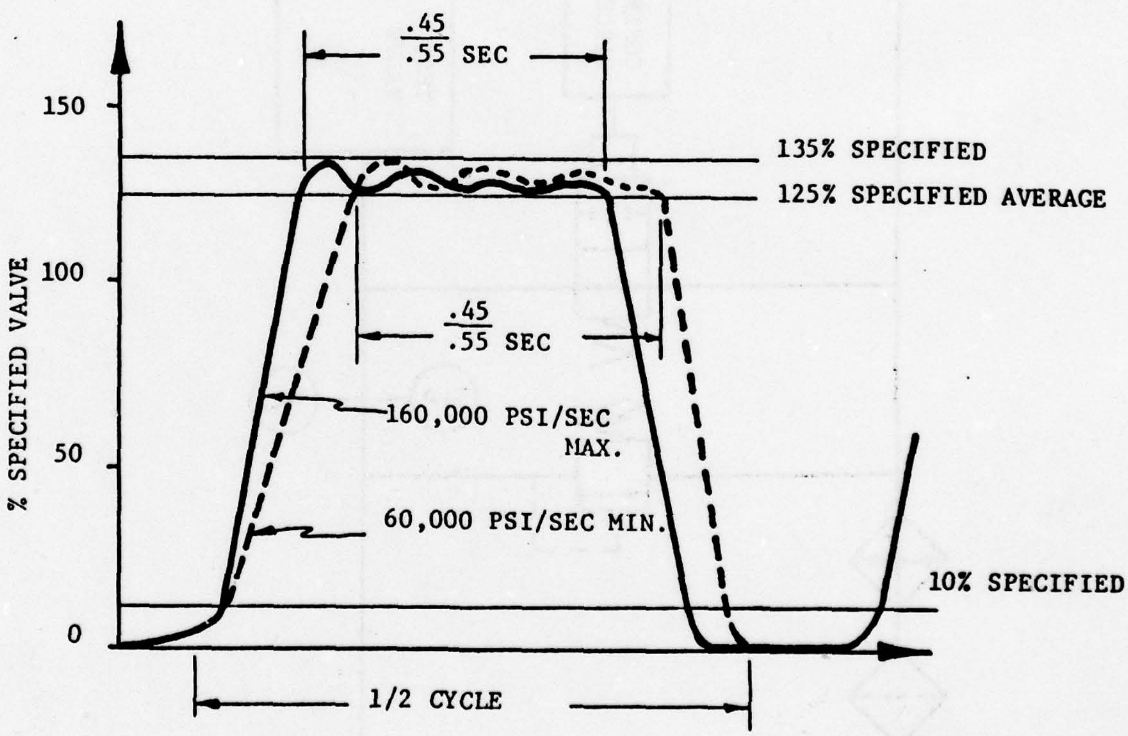


Fig. A-2. Inlet Pressure Cycle Associated with Durability Test.

8.5 Insure that the inlet pressure shall reach a minimum level of 10% of the rated value during each cycle and remain at below that level for at least 0.45 sec.

8.6 Adjust the time-rate-of-change-of-pressure not to exceed 160,000 PSI per second but to attain 60,000 PSI per second during pressure increases.

8.7 Measure and/or note and record all external leakage which causes drops to form within five minutes.

8.8 Clean the valve areas that are susceptible to external leakage at the completion of the total cycles.

8.9 Cycle the valve 1000 cycles, measure and/or note all external leakage which occurs during these cycles.

9. Data Presentation - Use Fig. A-3 as an example of a chart for test data.

10. Standardized Values.

10.1 A fluid with a viscosity of 21-26 mm^2/S at 50°C (122°F).

10.2 A test fluid temperature of 66°C (150°F)

11. Identification Statement - Use the following statement in catalogs and sales literature when electing to comply with this voluntary standard: "Performance data obtained and presented in accordance with SAE Recommended Practice J_____________."

12. References.

12.1 American National Standard Glossary of Terms for Fluid Power, ANSI/B93.2-1971.

Date Tested: ____________ Test Agency: ____________

Valve: ____________ Specified Flow: ____________

Fluid: ____________ Specified Pressure: ____________

Temperature: ____________ (°C) Specified Cycles: ____________

Viscosity: ____________ (cSt) Internally or Externally Mounted: ____

Maximum Specified Input: ____ Comments: ____________

Valve Readjusted at 50% Specified Cycles: ____________

LEAKAGE DURING TEST

LOCATION OF LEAKAGE	LEAKAGE RATE (ℓ/min)	CYCLES

LEAKAGE DURING FINAL 1000 CYCLES

LOCATION	LEAKAGE RATE (ℓ/min)

IF VALVE FAILED: Failure Location: ____________

No. Cycles To Failure: ____________

Fig. A-3. Example of a Chart for the Test Data.

12.2 International Standard Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmisson, ISO/R 1219-1970.

12.3 American National Standard Fluid Power Diagrams, ANSI/Y 14.17-1966.

12.4 American National Standard Graphic Symbols for Fluid Power Diagrams, Y32.10-1967.

12.5 International Standard Rules for the Use of Units of the International System of Units and a Selection of the Decimal Multiples and Sub-Multiples of SI Units, ISO/R 1000-1969.

METHOD FOR EVALUATING THE LOW TEMPERATURE PERFORMANCE OF A HYDRAULIC FLUID POWER VALVE - SAE XJXXX

1. Purpose - To provide a uniform procedure for measuring and reporting the low temperature operating characteristics of a hydraulic fluid power valve in order to insure that the valve continues to perform reasonably well in a system exposed to an extremely cold environment.

2. Terms and Definitions - (For definitions of terms not herein defined, see Ref. [1].).

   2.1 Specified Flow - Steady-state flow rate for the component as specified by the manufacturer.

   2.2 Specified Pressure - Steady-state operating pressure for the component as specified by the manufacturer.

   2.3 Control Variable - Any variable which causes the controlled output characteristic of the valve to change.

3. Units

   3.1 The international System of Units (SI) is used herein in accordance with Ref. [2].

   3.2 Approximate conversion to SI units appear in parentheses after their "Customary U.S." counterpart.

4. Graphic Symbols - Graphic Symbols used herein are in accordance with Refs. [3], [4], and [5]. Where Refs. [4] and [5] are not in agreement with Ref. [3], Ref. [3] governs.

5. Summary of Designated Information

   5.1 Specify the following information on all requests for this test:

5.1.1 A description of valve.

5.1.2 A description of fluid (if different from paragraph 10.1).

5.1.3 The operating fluid temperature.

5.1.4 Test pressure.

5.1.5 Test flow rate.

5.1.6 The input port (A).

5.1.7 The output port (A).

5.1.8 The work port (A).

5.1.9 The control variable.

6. GENERAL PROCEDURE

6.1 Conduct the test in accordance with the fixed values specified by the test request.

6.2 Use only standardized values, shown in paragraph 10.0, for catalog information and sales literature.

6.3 Test and report of subplate mounted valves may be run with the subplate included. Reports shall explicity indicate how such valves were tested.

7. TEST CONDITIONS

7.1 Accuracy - Maintain the test condition accuracy with the limits shown in the following table:

Test Condition	Maintain Within I
Flow	2%
Pressure	2%
Temperature	3^{o}C (5^{o}F)
Force	5%
Mechanical Displacement	2%
Voltage	3%
Current	3%

7.2 Contaminant Level - Limit the number of particles in the system fluid to a maximum of 1000 particles per millilitre greater than 10 μm with "clean-up" filter.

7.3 Test system volume shall be numerically equal to one-fourth the specified flow per minute.

7.4 Test environment should be an ambient temperature of -30^{o}C.

8. Test Procedure

8.1 Install the valve in the test circuit (Reference Fig. A-4).

8.2 Circulate system fluid through system and "clean-up" filter for fifteen minutes.

8.3 By-pass "clean-up" filter.

8.4 Set test system relief at 1.5 the test valve specified pressure at specified flow through the relief valve.

8.5 Lower the temperature of the entire test system to -30^{o}C for at least twelve hours.

8.6 Start the system flow.

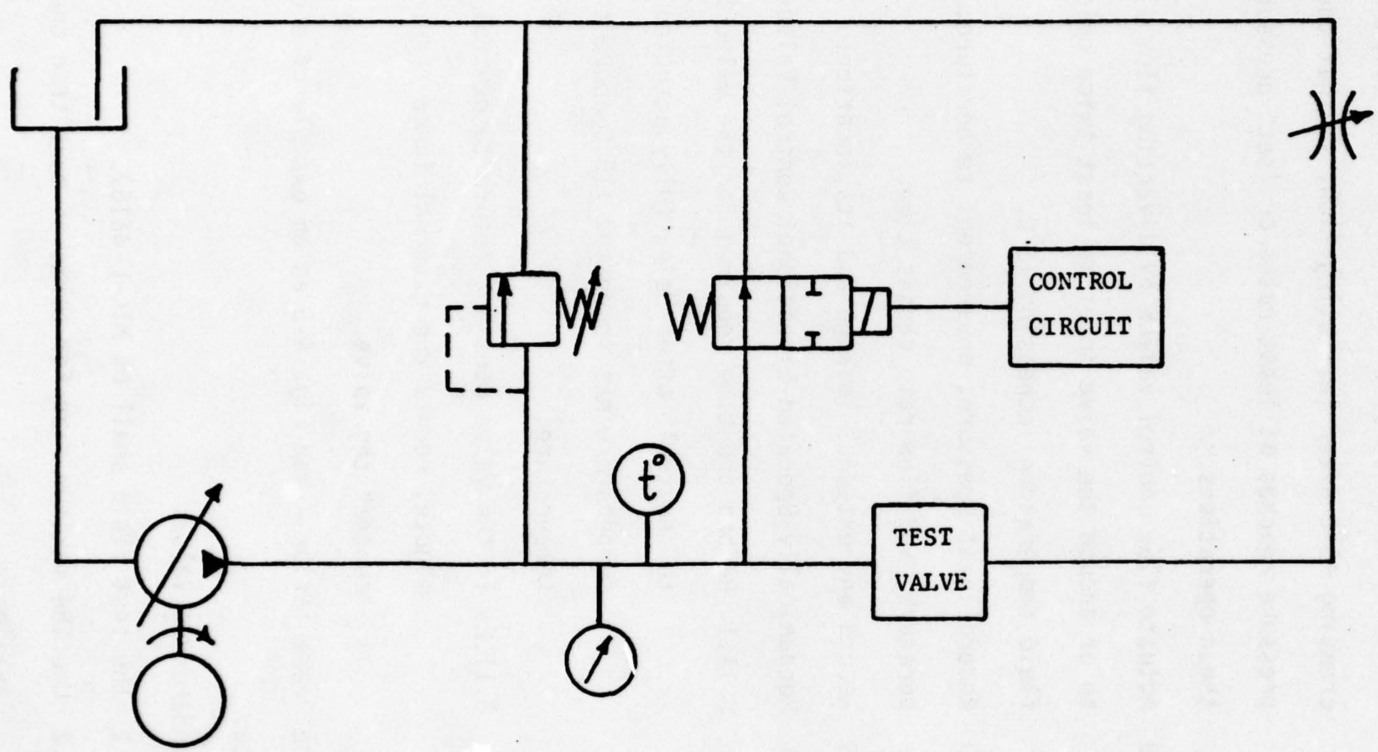


Fig. A-4. Example Fluid Power Test System for Low Temperature Test.

8.7 Actuate directional control valve or pressure control valves at least twice prior to the fluid temperature increasing +5°C after initiating flow, so that the inlet pressure reaches at least rated or "set" pressure during these operations.

8.8 Actuate flow control valves by diverting flow alternately to or around the valve inlet at least twice prior to the fluid temperature increasing +5°C.

8.9 Record inlet pressure, environment temperature, fluid temperature, and flow rate versus time.

8.10 Record any external leakage and its location.

8.11 Mechanically Operated Directional Control Valves.

8.11.1 Record the time required for the valve to return to "neutral" after being fully actuated when the temperature has increased +15°C above the test temperature.

8.11.2 If the valve does not return to neutral within one minute, record the external force required to "re-center" the valve.

9. Data Presentation - Use Fig. A-5 as an example of a chart for test data.

10. Standardized Values

10.1 The test fluid shall be MIL-L-46167.

10.2 Use the maximum size for ports if more than one size is available.

Fig. A-5. EXAMPLE LOW TEMPERATURE TEST DATA.

Date Tested: ________________ Test Location: ________________

Valve: ________________ Specified Flow: ________________

Fluid: ________________ Specified Pressure: ________________

Test System Volume: ________________ Comments: ________________

Test Temperature: $-30^{\circ}C$. ________________

ENVIRONMENT COOLING & STABILIZATION

Time	Temperature $^{\circ}C$

VALVE TEST

Inlet Pressure (bar)	System Flow (ℓ/min)	Environment Temperature $^{\circ}C$	Fluid Temperature $^{\circ}C$	Time	Leakage (ℓ/min) & Location

Mechanically Operated
Directional Control Valve

1. Time to Return to Neutral After Activation: ____________(sec)
(Fluid Temperature $-15^{\circ}C \pm 6^{\circ}C$.)

2. Force Required to "Recenter" After: ________________(N)
One Minute Waiting Period

11. Identification Statement - Use the following statement in catalogs and sales literature when electing to comply with this voluntary standard: "Performance data obtained and presented in accordance with SAE Recommended Practice JXXXX."

12. References

12.1 American National Standard Glossary of Terms for Fluid Power, ANSI/B93.2-1971.

12.2 International Standard Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmission, ISO/R 1219-1970.

12.3 American National Standard Fluid Power Diagrams, ANSI/y 14.17-1966.

12.4 American National Standard Graphic Symbols for Fluid Power Diagrams, Y32.10-1967.

12.5 International Standard Rules for the Use of Units of the International System of Units and a Selection of the Decimal Multiples and Sub-Multiples of SI Units, ISO/R 1000-1969.

METHOD OF EVALUATING THE STRUCTURAL

INTEGRITY OF A HYDRAULIC FLUID POWER

VALVE - SAE JXXXX

1. Purpose - To provide a uniform procedure for measuring and reporting an apparent structural integrity of a hydraulic fluid power valve.

2. Terms and Definitions - (For definitions of terms not herein defined, see Ref. [1]).

   2.1 Specified Pressure - Steady-state operating pressure for the appropriate ports as specified by the manufacturer.

   2.2 Input Port - Any port into which flow is directed as specified by the manufacturer.

   2.3 Control Variable - Any input variable which causes the controlled output characteristic of the valve to change.

   2.4 Output Port - Any port which is subjected to high pressure fluid while conducting modulated flow to another component.

3. Units

   3.1 The International System of Units (SI) is used herein in accordance with Ref. [2].

   3.2 Approximate conversion to SI units appear in parentheses after their "Customary U.S." counterpart.

4. Graphic Symbols - Graphic symbols used herein are in accordance with Refs. [3], [4], and [5]. Where Refs. [4] and [5] are not in agreement with [3], Ref. [3] governs.

5. Summary of Designated Information

5.1 Specify the following information on all requests for this test:

5.1-1 A description of valve.

5.1-2 A description of fluid (if different from paragraph 10.1)

5.1-3 The fluid temperature (if different from the standardized valve in paragraph 10.2).

5.1-4 Test pressure.

5.1-5 The input port(s).

5.1-6 The output port(s).

5.1-7 The work port(s).

5.1-8 The control variable position(s).

6. General Procedure

6.1 Conduct the test in accordance with the fixed values specified by the test request.

6.2 Use only standardized values, shown in paragraph 10.0, for catalog information and sales literature.

6.3 Test and report of subplate mounted valves may be run with the subplate included. Reports shall explicitly indicate how such valves were tested.

7. Test Conditions

7.1 Accuracy - Maintain the test condition accuracy within the limits shown in the following table:

Test Condition	Maintain Within ±
Flow	2%
Pressure	2%
Temperature	$3^{o}C$ ($5^{o}F$)
Control Variable	2%

7.2 Contaminant Level - Limit the number of particles in the system fluid to a maximum of 1000 particles for millilitre greater than 10 μm.

8. Test Procedure

8.1 Install the valve in the test circuit (Reference Fig. A-6) so that high pressure fluid may be directed to the inlet and all work ports.

8.2 Connect the outlet to the tank.

8.3 Pressurize the inlet to two times the specified inlet pressure for at least 60 seconds. (This test does not apply to relief valves.)

8.4 Pressurize the outlet(s) to two times the specified return pressure for at least 60 seconds.

8.5 Pressurize the work ports to two times the specified work port pressure for at least 60 seconds.

9. Data Presentation - Use Fig. A-7 as an example of a chart for test data.

10. Standardized Valves

10.1 A fluid with a viscosity of 21-26 mm^2/s at $50^{o}C$ (105-125 SUS at $122^{o}F$) and 6.6-7.4 mm^2/s at $90^{o}C$ (48-50 SUS at $194^{o}F$) should be used.

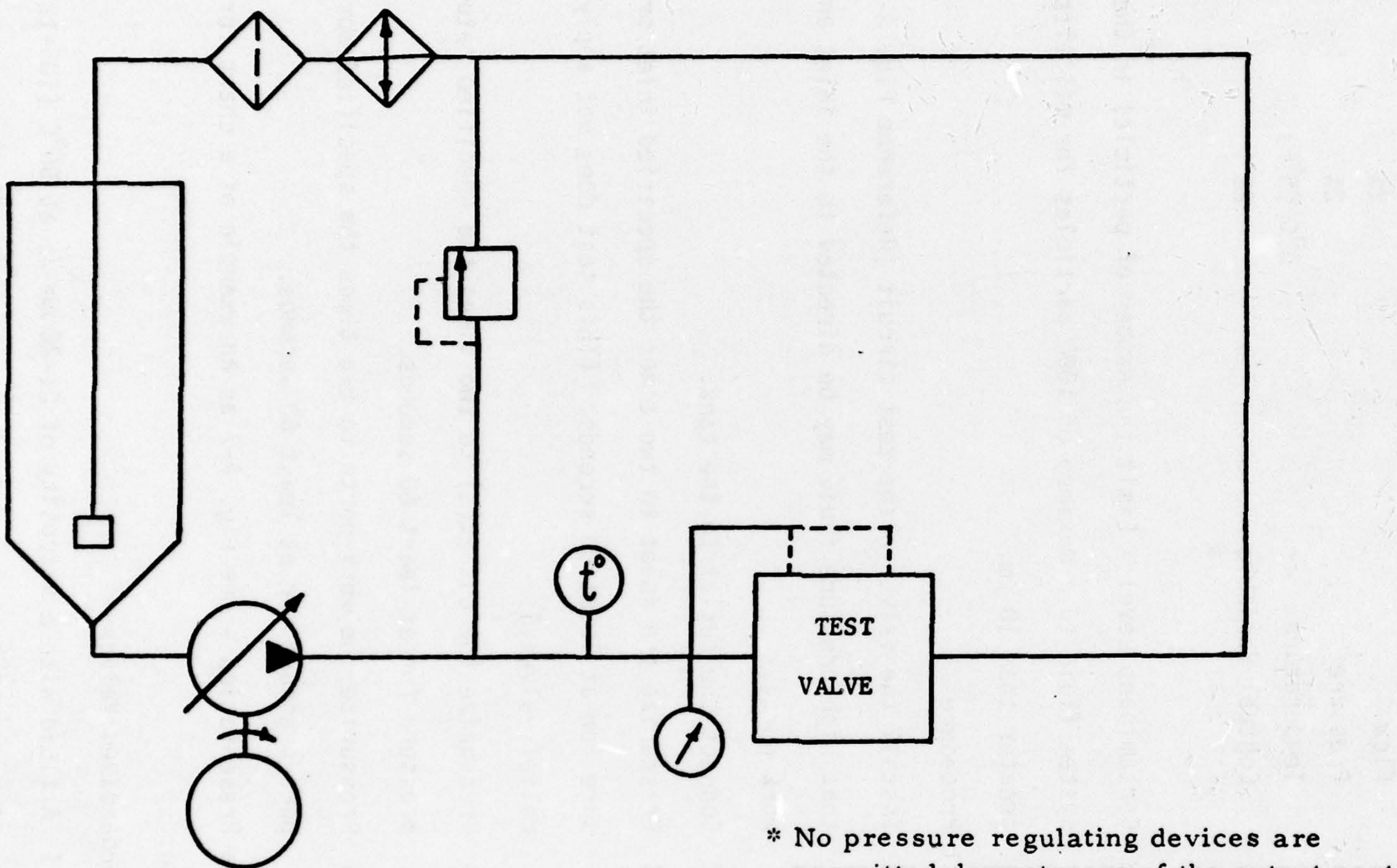


Fig. A-6. Example Fluid Power Test System for Evaluating the Structural Integrity of a Valve.

STRUCTURAL INTEGRITY TEST

VALVE DESCRIPTION ______________________________

TEST AGENCY ______________________ DATE TESTED __________

TYPE OF FLUID ______________________ FLUID TEMP. __________

FLUID VISCOSITY ______________________ CONTROL POSITION __________

SPECIFIED INPUT PRESSURE: ______________________ COMMENT: __________

SPECIFIED OUTPUT PRESSURE: ______________________

SPECIFIED WORK PORT PRESSURE: ______________________

PORT TYPE AND LOCATION	TEST PRESSURE	LEAKAGE RATE AND LOCATION

Fig. A-7. Example of a Chart for the Test Data.

10.2 Fluid temperature should be $66^{\circ}C$ ($150^{\circ}F$).

11. Identification Statement - Use the following statement in catalogs and sales literature when electing to comply with this voluntary standard: "Performance data obtained and presented in accordance with SAE Recommended Practice JXXXX."

12. References

12.1 American National Standard Glossary of Terms for Fluid Power, ANSI/B93.2-1971.

12.2 International Standard Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmission, ISO/R 1219-1970.

12.3 American National Standard Fluid Power Diagrams, ANSI/Y 14.17-1966.

12.4 American National Standard Graphic Symbols for Fluid Power Diagrams, Y32.10-1967.

12.5 International Standard Rules for the use of units of the International System of Units and a Selection of the Decimal Multiples and Sub-Multiples of SI Units, ISO/R 1000-1969.

SALT SPRAY TEST METHOD FOR HYDRAULIC
FLUID POWER VALVE-SAE JXXXX — SAE Recommended Practice

Report of Off-Road Machinery Technical Committee approved ______________

1. Purpose

To provide a uniform procedure to perform a salt spray test on a fluid power valve.

2. (Terms and Definitions)

2.1 Fog Chamber - A chamber in which a test valve is exposed to salt spray (fog).

2.2 Compressed air supply - A supply of suitably conditioned compressed air that sprays salt solution into the fog chamber to create salt fog continuously.

2.3 Atomizing nozzle - Any nozzle which can create a salt fog by mixing compressed air and salt/water solution.

3. Units

3.1 The International System of Units (SI) is used herein in accordance with Reference [1].

3.2 Approximate conversion to SI units appear in parenthesis after their "Customary U.S." counterpart.

4. Summary of Designated Information

4.1 Specify the following information on all requests for this test:

4.1.1 A description of test valve.

4.1.2 A description of salt solution (if different from paragraph 8.1).

4.1.3 Fog chamber temperature (if different than paragraph 6.2).

4.1.4 Test period (if different from paragraph 7.5).

5. General Procedure

5.1 Conduct the test in accordance with the fixed values by the test request.

5.2 Use only standardized values, shown in paragraph 8, for catalog information and sales literature.

5.3 Test and report of subplate mounted valves may be made with the subplate included. Reports shall explicitly indicate how such valves were tested.

6. Test Conditions

6.1 Position the valve in the fog chamber during the test such that the following condition is met:

6.1.1 The valve shall be tilted and supported between 15 and 30 degrees from the vertical and preferably parallel to the principal direction of horizontal flow of the fog through the chamber, based upon the dominant surface being tested.

6.2 Temperature in the fog chamber is 95 + 2, -3°F (35 + 1.1, -1.7°C).

6.3 Fog chamber and atomizing nozzles shall be designed to distribute the salt fog uniformly throughout the chamber.

6.4 Atomizing nozzle or nozzles shall be directed or baffled such that none of the spray can impinge directly on the valve.

7. Test Procedure

7.1 Plug all ports of the valve completely.

7.2 Clean the valve. No dirt or oil should be visible on the surface of valve.

7.3 Install the valve in the fog chamber as specified in paragraph 6.1.1.

7.4 Start the test with the temperature specified in paragraph 6.2.

7.5 The period of test shall be 30 hours.

7.6 After the termination of test, the valve shall be gently cleaned by an oily rag.

7.7 Upon completion examine the valve. Any staining or corrosion that cannot be removed by rubbing with an oily rag or evidence of pitting shall be recorded and reported as a failure of this test.

8. Standardized Values

8.1 The salt solution shall be prepared by dissolving 5 ± 1 parts by weight of sodium chloride in 95 parts of distilled water or water containing not more than 220 ppm of total solids.

8.2 The compressed air supply to the nozzle or nozzles for atomizing the salt solution shall be free of oil and dirt and maintained between 10 and 25 psi (69 and 172 kN/m^2).

8.3 The fluid temperature of 150°F (66°C).

9. Identification Statement - Use the following statement in catalogs and sales literature when electing to comply with this voluntary standard: "Performance data obtained and presented in accordance with SAE Recommended Practice JXXXX."

10. References

10.1 International Standard Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmission, ISO/R 1219-1970.

METHOD OF MEASURING AND REPORTING
THE CONTAMINANT LOCK CHARACTERISTICS
OF A HYDRAULIC FLUID POWER,
VALVE-SAE______________________ SAE Recommended Practice

Report of Off-Road Machinery Technical Committee Approved____________

1. Purpose

To provide a uniform procedure for measuring and reporting the contaminant lock characteristics of a hydraulic fluid power valve.

2. (Terms and Definitions)

(For definitions of terms not herein defined, see References).

2.1 Contaminant - Any foreign substance that can have deleterious effects on system operation.

2.2 Contaminant lock - Any situation where the valve spool or activating mechanism is restrained or locked due to contaminant interference.

2.3 Initial friction force - Initial input force required to shift the valve spool.

2.4 Stationarity time - The time period from the contaminant injection to the time spool operation or movement is attempted.

2.5 Silting Force - The additional force needed to shift the valve spool after exposure to a contaminant environment.

2.6 Gravimetric level - Contaminant weight per unit fluid volume.

2.7 Upper Cut Dust - Classified AC Fine Test Dust for the range greater than a specified value.

3. Units

3.1 The International System of Units (SI) is used herein in accordance with Reference Paragraph 12.2.

3.2 Approximate conversion to SI units appear in parentheses after their "Customary U.S." counterpart.

4. Graphic Symbols

Graphic symbols used herein are in accordance with References (paragraphs 12.3 and 12.4.). Where References [12.3] and [12.4] are not in agreement, Reference [12.3] governs.

5. Summary of Designated Information

5.1 Specify the following information on all requests for this test:

5.1.1 A full description of the valve.

5.1.2 The type of fluid (if different than clause 10.1).

5.1.3 The fluid temperature (if different from the standardized values in clause 10.2).

5.1.4 The test pressure (if different than clause 10.3.)

5.1.5 The test flow rate.

5.1.6 The test contaminant (if different than clause 10.4.).

6. General Procedure

6.1 Conduct the test in accordance with the operating parameters specified in the test request.

6.2 Use only standardized operating valves, shown in clause 10.0, from catalog information and/or from sales literature.

7. Test Condition Accuracy

Maintain the test condition accuracy within the limits shown in the following table:

Test Condition	Maintain Within ±
Flow	2%
Pressure	2%
Force	5%
Mechanical Displacement	2%
Temperature	$3^{o}C$ ($5^{o}F$)

8. Test Procedure

8.1 Install the test valve in the test circuit (Reference Fig. A-8).

8.2 Operate the test system with the filter so that the contamination level is less than 1000 particles greater than 10 μm per mℓ

8.3 Set the operating parameters as specified.

8.4 Measure the initial actuation friction or spool force.

8.5 Valve out the filter from the system.

8.6 Inject the classified AC Fine Test Dust of 0-5 μm 50 mg/ℓ into the system.

8.7 Measure the silting forces for the stationarity times of 4, 8, and 16 minutes.

8.8 Connect the filter to the system and reduce the contamination level of the fluid to less than 1000 particles greater than 10 μm/mℓ.

8.9 Repeat clause 8.4 to 8.8 for 50 mg/ℓ of upper cut dust of the following sizes: 5μm UP, 10μm UP, 20μm UP, 30μm UP, and 40μm UP.

9. Data Presentation

9.1 Tabulate the test data.

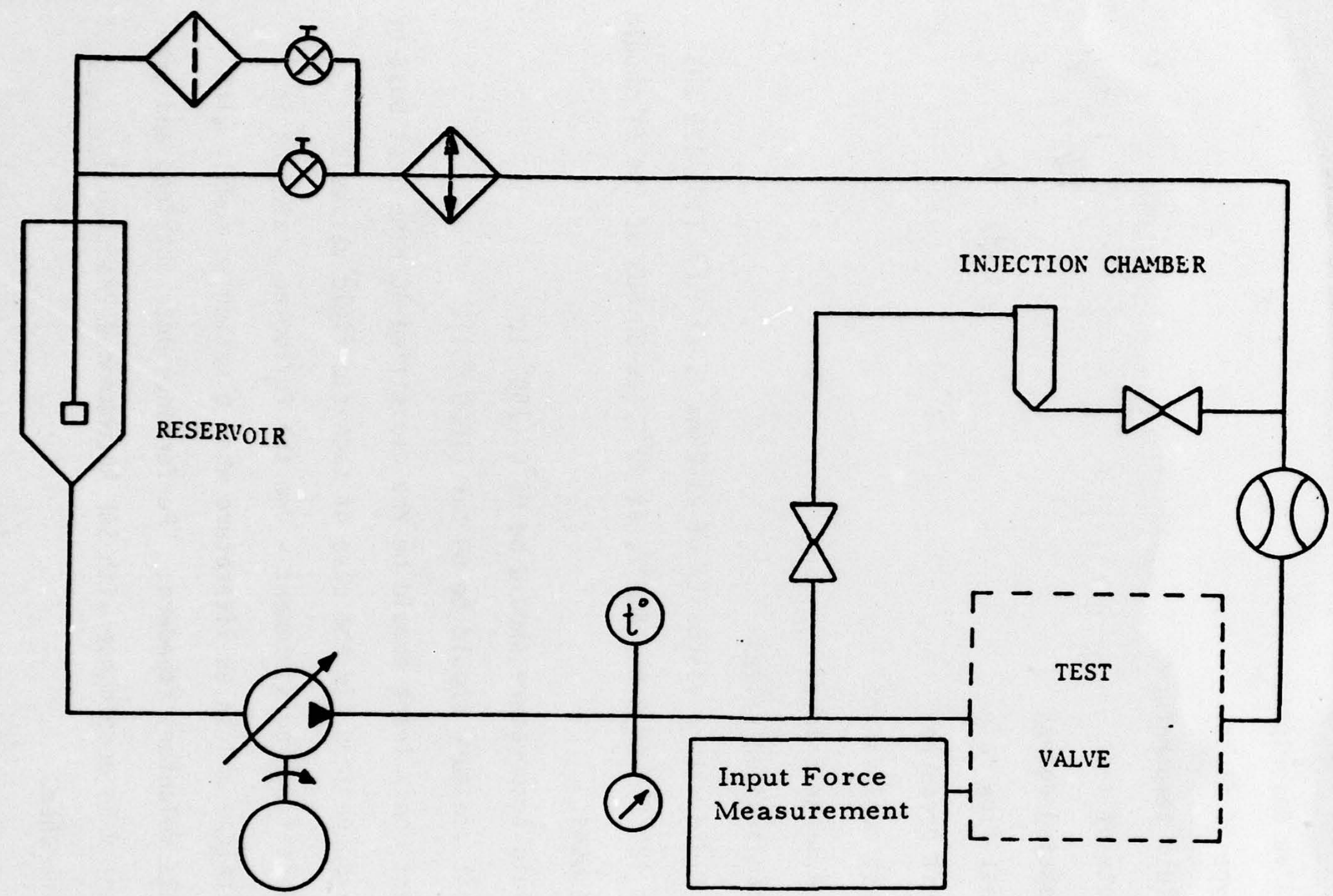


Fig. A-8. Schematic Example of a Hydraulic Valve Test System.

9.2 Use Fig. A-9 as an example table.

9.3 Include the following information on the data table:

9.3.1 Valve description

9.3.2 Type of fluid

9.3.3 Fluid temperature

9.3.4 Date of test

9.3.5 Testing agency

9.3.6 Test flow rate

9.3.7 Test pressure

9.3.8 Test contaminant

9.3.9 Gravimetric level

10. Standardized valves

10.1 A fluid with a viscosity of 21-26mm^2/s at 50°C (105-125 SUS at 122°F) and 6.6-7.4mm^2/s at 90°C (48-50 SUS at 194°F) should be used.

10.2 Fluid temperature should be 65°C (150°F).

10.3 Test pressure should be 69 bar (1000 PSI).

10.4 Test contaminant should be the classified AC Fine Test Dust or Carbonye Iron (in the case of magnetic field valves).

11. Identification Statement - Use the following statement in catalogs and sales literature when electing to comply with this voluntary standard: "Performance data obtained and presented in accordance with SAE Recommended Practice J_______."

12. References

12.1 American National Standard Glossary of Terms for Fluid Power, ANSI/B93.2-1971.

TEST REPORT SHEET

VALVE DESCRIPTION ______________________________

TEST AGENCY ______________________ DATE ____________

TYPE OF FLUID ______________ FLUID TEMP. ____________

TYPE OF CONTAMINANT ____________________________

GRAVIMETRIC LEVEL ______________________________

TEST PRESSURE ______________ TEST FLOW RATE __________

COMMENTS ______________________________________

____

CONTAMINANT SIZE (μm)	MEASURED VALUE				SILTING FORCE		
	A 0 min	B 4 min	C 8 min	D 16 min	B-A 4 min	C-A 8 min	D-A 16 min
0-5							
5-UP							
10-UP							
20-UP							
30-UP							
40-UP							

Fig. A-9. Example of a Table for the Test Data.

12.2 International Standard Graphical Symbols for Hydraulic and Pneumatic Equipment and accessories for Fluid Power Transmission, ISO/R 1219-1970.

12.3 American National Standard Fluid Power Diagrams, ANSI/Y 14-17-1966.

12.4 American National Standard Graphic Symbols for Fluid Power Diagrams, Y32.10-1967.

12.5 International Standard Rules for the Use of Units of the International System of Units and a Selection of the Decimal Multiples and Sub-Multiples of SI Units, ISO/R 1000-1969.

ABRASIVE TEST FOR A HYDRAULIC
FLUID POWER VALVE-SAE____________ SAE Recommended Practice

Report of Off-Road Machinery Technical Committee approved____________

1. Purpose - To provide a uniform procedure to conduct an abrasive test on a hydraulic fluid power valve.

2. Terms and Definitions

(For definitions of terms not herein defined, see Reference [1]).

2.1 Specified Flow - Steady-state flow rate for the valve as specified by the manufacturer.

2.2 Specified Pressure - Steady-state operating pressure for the valve as specified by the manufacturer.

2.3 Abrasive Environment - A circulating air-dust zero visibility consisting of 30-45 mg/ft^3 (1.0-1.5 g/m^3) of AC Coarse Test Dust.

2.4 AC Coarse Test Dust - The test dust specified in SAE J7266.

3. Units

3.1 The International System of Units (SI) is used herein in accordance with Reference [2].

3.2 Approximate conversion to SI units appear in parentheses after their "Customary U.S." counterpart.

4. Graphic Symbols - Graphic symbols used herein are in accordance with References [3], [4], and [5]. Where References [4] and [5] are not in agreement with [3], Reference [3] governs.

5. Summary of Designated Information

5.1 Specify the following information on all requests for this test:

5.1.1 A description of valve.

5.1.2 A description of fluid (if different from paragraph 10.1).

5.1.3 The fluid temperature (if different from the standard value in paragraph 10.2).

5.1.4 Test flow.

5.1.5 Test pressure.

5.1.6 Test dust (if different from the paragraph 7.4).

5.1.7 The test environment temperature (if different from the paragraph 10.3).

5.1.8 The abrasive environment (if different from the paragraph 2.3).

5.1.9 Test period (if different from the paragraph 9.6).

5.1.10 A rate of cycles of the best value operation (if different from the paragraph 9.4).

6. General Procedure

6.1 Conduct the test in accordance with the fixed values specified by the test request.

6.2 Use only standardized values, shown in paragraph 10, for catalog information and sales literature.

6.3 Test and report of subplate mounted valves may be run with the subplate included. Reports shall explicitly indicate how such values were tested.

7. Test Conditions

7.1 Accuracy - Maintain the test condition accuracy within the limits shown in the following table:

Test Condition	Maintain Within ±
Flow	2%
Pressure	2%
Temperature	$3^{\circ}C$ ($5^{\circ}F$)
Humidity	5%

7.2 Initial contaminant Level of Fluid - Limit the number of particles in the system fluid to a maximum of 1000 particles per millilitre greater than 10 micrometres with "clean-up" filter.

7.3 Test system volume shall be numerically equal to one-fourth the specified flow per minute.

7.4 AC Coarse Test Dust shall be used to create the abrasive environment.

8. Abrasive Environment Qualification Procedure

8.1 Establish the proposed (zero visibility) air-dust environment using AC Coarse Test Dust in the chamber.

8.2 Sample (twice over a 4-hour period) the air-dust environment by drawing a known volume of air-dust mixture through a pre-weighed 0.45 μm membrane.

8.3 Weigh the membranes obtained in paragraph 8.2 and calculate the weight of dust in the unit volume.

8.4 Consider the abrasive environment qualified if the two gravimetric levels obtained in paragraph 8.3 meet the requirement specified in paragraph 2.3.

9. Test procedure

9.1 Install the test valve in the test circuit as illustrated in Figure A-10.

9.2 Operate the test system at the specified operating parameters in a clean environment with the filter in the system such that the fluid contaminant level is lower than that specified for the initial condition of the test.

9.3 Isolate the filter from the test system and expose the test valve to the abrasive environment.

9.4 Reciprocate the actuator of the test valve at a rate of 20 cycles per minute.

9.5 Maintain all test conditions as specified throughout the test.

9.6 Continue the test for 10 hours.

9.7 After the test is completed, examine the test valve for evidence of wear, damage, and external leakage. Any such evidence found on the valve shall be reported as a failure.

9.8 Upon completion of the above procedure, the test valve shall be tested as specified in SAE J1117 and SAE J1235. Inability of the test valve to satisfy the above specified tests shall constitute failure of this test.

10. Standardized values

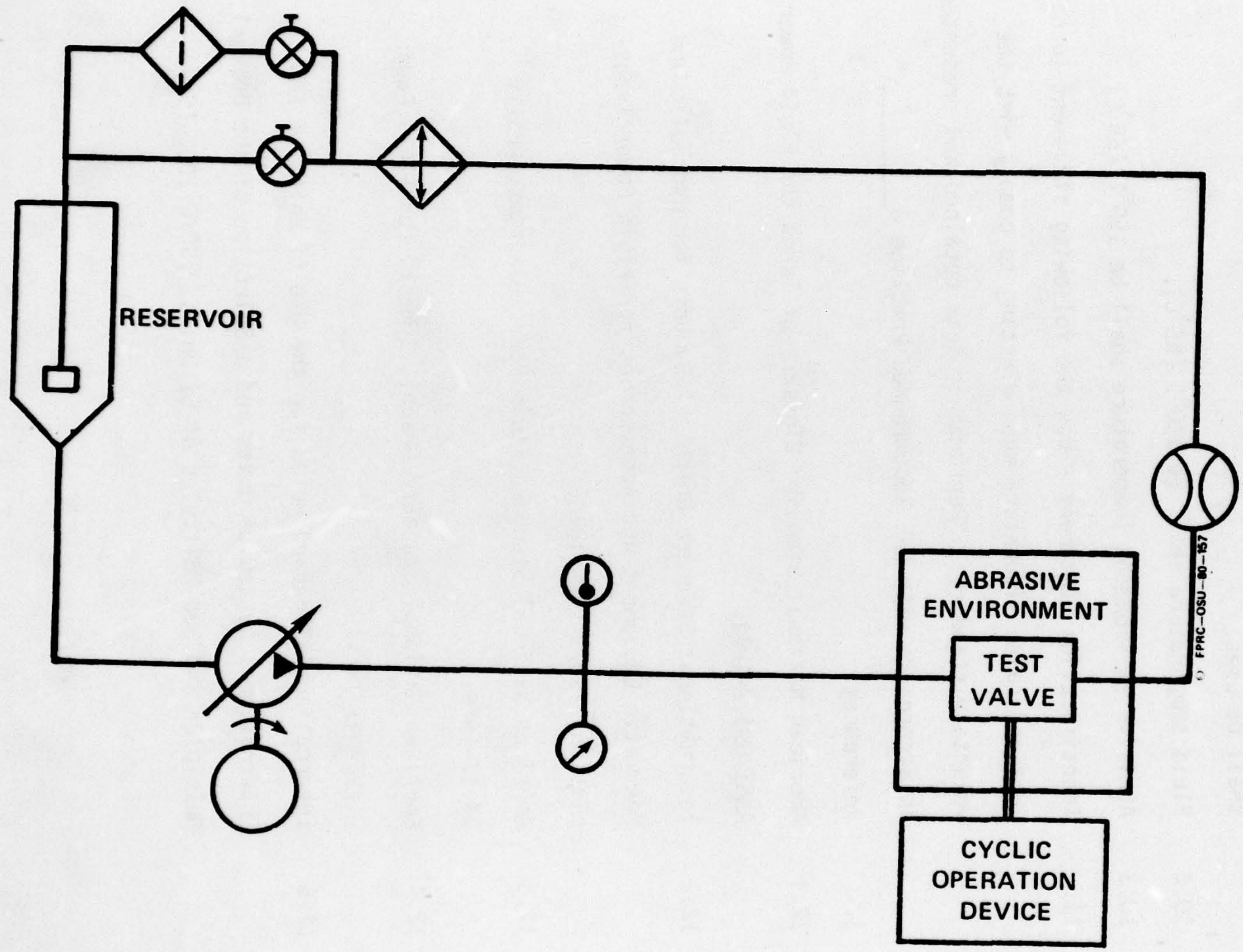


Fig. A-10. Schematic Example of a Hydraulic Valve Test System.

10.1 A fluid with a viscosity of 21-26 $mm^2/_s$ at 50°C (105-125 SUS at 122°F) and 6.607.4 $mm^2/_s$ at 90°C (48-50 SUS at 194°F) shall be used.

10.2 Fluid temperature shall be 150°F (65°C).

10.3 Abrasive Environment Temperature shall be 150°F (65°C).

11. Identification Statement - Use the following statement in catalogs and sales literature when electing to comply with the voluntary standard: "Performance data obtained and presented in accordance with SAE Recommended Practice J________."

12. References

12.1 American National Standard Glossary of Terms for Fluid Power, ANSI/B93.2-1971.

12.2 International Standard Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmission, ISO/R 1219-1970.

12.3 American National Standard Fluid Power Diagrams, ANSI/y 14.17-1966.

12.4 American National Standard Graphic Symbols for Fluid Power Diagrams, Y32.10-1967.

12.5 International Standard Rules for the Use of Units of the International System of Units and a Selection of the Decimal Multiples and Sub-Multiples of SI Units, ISO/R 1000-1969.